POWER

Volume 55 New York, March 28, 1922 *Number* 13

Time

THERE is a great deal said about the inequality of opportunity, but there is one thing of which we all have an equal amount, and that is Time.

And Time is Money.

What one will be ten years from now depends upon how he spends those ten years.

And just as with money, if you take care of the pennies the dollars will take care of themselves; if you take care of and spend to advantage the odd minutes, the hours and the years will be full of accumulated knowledge and accomplishment.

The man who has no time to read fritters away time on the train skimming column after column of newspaper matter that he will forget before he gets to his shop or office. The same time would have sufficed for the reading of a chapter or two of some good book that would have made him a better man in his business or a better citizen of his community.

I do not mean that one should not keep in touch with affairs by acquainting himself with the daily news. But the really important happenings of the ordinary day, the things that really make much difference, are few. It is not absorbing these and the editorials and thoughts of others concerning them that takes the most of the thought and time of the daily-paper reader.

As it is interest that makes money grow and yield, so in the case of time it is interest that leads to profitable investment and worth-while returns. If a man is interested enough in a subject, he will find time to pursue it. If one has a hobby, one is never without time to ride it. And success depends upon the selection of a hobby, the cultivation and inciting of interest in worthwhile things that knowledge of and proficiency in will get you somewhere; and then economizing your time and devoting the minutes and hours that others fritter away to these objects the accomplishment and mastery of which will mean progress and success.

Heat Balance and Steam Distribution in a Large Service Plant

Tells How Steam, Hot Water and Power Are Charged to Different Departments in a Large Building Supplied from an Isolated Power Plant

By S. D. KUTNER

ALMOST every operating engineer has been faced with the problem of determining the amount of steam that the various departments in an industry should be charged with. The purpose of this article is to tell how to develop a heat balance and a steam distribution record for a service plant supplying power and heat for a number of purposes. The plant produces steam which is utilized for: (1) Producing compressed air by means of two-stage air compressors; (2) operating hydraulic-elevator pumps; (3) operating a fire-protection system; (4) pumping cold water used as plumbing supply; (5) heating water for plumbing and washroom service; (6) heating the water used in a hot-water heating system; (7) operating turbo-generators during the heating season; (8) supplying high-pressure steam for heating buildings; (9) supplying high-pressure steam for domestic and miscellaneous use.

The following list shows the steam producing and consuming apparatus in the plant as well as the electrically operated pumps:

Four hand-fired boilers, having 5,500 sq.ft. of heating surface.

Ten boilers, having 6,350 sq.ft of heating surface; 5 of which are equipped with chain-grate stokers and 5 are hand-fired.

One steam-driven boiler-feed pump.

Two turbine-driven boiler-feed pumps.

One motor-driven boiler-feed pump.

One feed-water heater, 240,000-lb. per hr. capacity.

Three tandem-compound duplex-piston hydraulic-elevator pumps.

Two motor-driven elevator pumps.

Two motor-driven, hot-water, plumbing-supply pumps.

One hot-water plumbing-supply heater.

One cold-water, plumbing-supply tandem-compound pump.

Two steam-driven underwriters fire pumps.

One motor-driven fire pump.

Eight motor-driven hot-water heating circulating pumps.

Three motor-driven hot-water heating-system condensation pumps.

Five hot-water heating-system heaters, 6,000 gal. per min.

One two-cylinder coal skip-hoist engine.

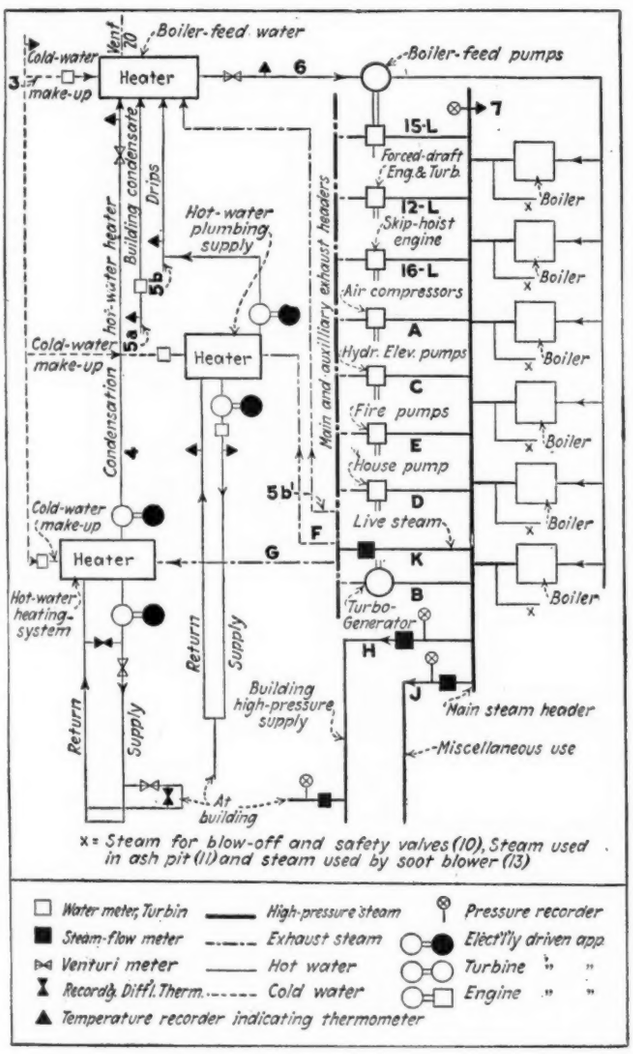
One vertical motor-driven hot-water drip pump.

Two forced-draft blower engines.

Two turbine-driven single-stage blowers.

Steam Available for Distribution

In the figure is given a diagrammatic layout of the plant, the numbers and letters on the diagram referring to the items on the forms for the monthly heat balance and distribution of steam in Tables I and II. Cold makeup water coming from the surge and storage tanks is fed to the boiler feed-water heater, the plumbing hot-water heater and to five hot-water heating-system heaters. The quantity and temperature of the feed water is measured as it is pumped to the boilers. If there are no leaks in the feed-water line, the quantity so measured is the gross pounds of steam produced, but in this plant steam is lost through the boiler blowoff, a steam



DIAGRAMMATIC LAYOUT OF PLANT EQUIPMENT

jet in the ashpit of each boiler used to break up the clinkers and the soot blowers. Of the heat in the steam used for operating the forced-draft equipment 20 per cent is also deducted.

Deducting the four items mentioned in the foregoing from the gross gives the net steam produced. Of the steam required to operate the boiler-feed pumps and coal

skip-hoist engine, 20 per cent is charged to the pumps and engine and the remainder to the exhaust. In addition, there is a vent loss in the heater and radiation loss in the plant piping.

As the exhaust steam from all pumps, turbines and engines is utilized in the heaters, it is the practice at this plant to charge 20 per cent of the heat energy required to operate the apparatus to work and 80 per cent to exhaust for heating purposes.

Monthly Heat Balance

Table I shows the heat balance for the plant as calculated for April, 1921. In the following each item is discussed separately:

1. The average heat value of the coal as fired is determined from an analysis of an average sample of the fuel burned during the month. This value is reduced to an "as fired basis" by taking into consideration the total moisture in the coal as fired. This total moisture is obtained from moisture tests made on ten-pound samples daily and is shown on the report covering the proximate analysis of the average sample of coal burned.

TABLE I—MONTHLY HEAT BALANCE

		April, 1921			
		A	B	C	D
		Miscl.	Temp.	Thous. Lbs.	Million B.t.u.
1	Average heat value of coal as fired, B.t.u.	10,548	.....	.....	.....
2	Total coal as fired	.....	.....	10,838	114,319
3	Water, makeup	.....	42.7	43,058	.....
4	Water, condensate, hot-water heating system	.....	189.5	13,493	1,981
5a	Water, condensate from buildings	.....	136.0	1,984	185
5b	Water, all other condensate	.....	115.7	8,834	8,566
6	Water, total boiler feed	.....	202.0	67,369	10,732
7	Steam, average gage pressure and average temperature	178.0	455.0	.....	.....
8	Steam, factor of evaporation	1.106	.....	.....	.....
9	Steam, total equivalent evaporation	.....	.....	74,533	72,327
10	Steam loss at blowoff and safety valves	.....	.....	.....	96
11	Steam used in ashp t	.....	.....	.....	694
12	Steam charged to forced draft (20 per cent)	.....	.....	.....	300
13	Steam used by soot blowers	.....	.....	.....	333
14	Steam, net produced [9—(10+11+12+13)]	.....	.....	.....	70,904
15	Steam, 20 per cent of amount used by boiler-feed pumps	.....	.....	.....	368
16	Steam, 20 per cent of amount used by skip-hoist engine	.....	.....	.....	332
17	Steam, loss in feed-water heater	.....	.....	.....	107
18	Steam, line losses between boilers and apparatus	.....	.....	.....	71
19	Steam available for distribution [14—(15+16+17+18)]	.....	.....	.....	70,026
20	Steam, equiv. elec. power for operating stokers	.....	.....	.....	39
21	Steam, equiv. elec. power for operating coal conveyo	.....	.....	.....	14
22	Steam, equiv. elec power for operating boiler-feed pump	63.2	.....	.....	0
23	Boiler efficiency (9÷2)	.....	.....	.....	.....
24	Over-all plant efficiency [19—(20+21+22)÷2]	61.3	.....	.....	.....

2. Total coal fired is determined from the measurement of the fuel in the bunkers on the first day of each month and the total coal received as determined from the skip-hoist coal scales. Reducing the tons of coal to pounds, then the product of pounds by the B.t.u. per pound gives the total B.t.u. in the coal as fired.

3. Boiler-feed makeup water is measured in cubic feet by a turbine-type water meter, and the temperature is taken at the surge tank. The quantity in pounds is obtained by the product of the cubic feet and the pounds per cubic foot corresponding to the temperature.

4. Water, condensate from the hot-water heating system, is measured in gallons by means of a venturi meter and the temperature recorded by a recording thermometer. Gallons are changed to pounds, using the pounds per gallon corresponding to the temperature of the condensate. The right-hand column *D* shows the heat returned by this condensate above the temperature of the makeup water and is calculated from the pounds of condensate and the difference in temperature between the condensate and makeup water.

5a. Water, condensate returned from buildings, is measured in cubic feet by a turbine-type water meter and the temperature recorded by means of a recording thermometer. The cubic feet are reduced to pounds and from the difference in temperature between the condensate and makeup water, the heat returned to the feed-water heater above the temperature of makeup is determined.

5b. Water, all other condensate, in this report it is a combination of high-pressure and hot-water drip returns plus the exhaust steam taken from the exhaust-steam header (Item 5*b'* in the figure). Neither of these quantities is measured although the temperature of the drip returns is recorded. The weight of water and B.t.u. for these quantities are obtained by subtracting from the total boiler feed, item 6, the sum of items 3, 4, and 5a, both for columns *C* and *D*. However, at the present time it is proposed to measure the amount of drip return, leaving the quantity of exhaust steam from the header as the only unmetered amount and this can be calculated.

6. Water, total boiler feed, is measured on a venturi meter and the temperature recorded by means of a recording thermometer, and is reduced to pounds by multiplying the cubic feet by the pounds per cubic foot corresponding to the temperature of the feed water. The total heat in the feed water above the temperature of makeup is calculated from the pounds of boiler feed and the difference in temperature between the boiler feed and the makeup water.

7. Average steam pressure is recorded by a recording pressure gage, and the temperature of the superheated steam is obtained from readings of an indicating thermometer, in the steam header, and are recorded on the boiler-room log.

8. Factor of evaporation is calculated by deducting from the total heat in the steam at boiler conditions the total heat in the feed water above 32 deg. F. and dividing the result by 970.4.

9. Steam, total equivalent evaporation, in pounds, is obtained by multiplying the actual evaporation by the factor of evaporation, and this result, when multiplied by 970.4, gives the total equivalent evaporation from and at 212 deg. F. in B.t.u. This is the heat added to the feed water within the boiler proper.

10. Steam, loss at blowoff and safety valves: The blowoff is calculated from the number and size of the blowoff lines and the time the valves are kept open. For normal operation this has been reduced to pounds per boiler hour, and from the B.t.u. per pound of steam as supplied by the boiler, the total B.t.u. charged to blowoff is calculated. The safety valves rarely pop as a warning whistle gives an indication of high pressure in time to prevent them popping. In case they do, however, a record of the time they are open is reported and the steam so lost is calculated, knowing the size of the valves.

11. Steam used in the ashpit: A small steam jet is used continuously while each boiler is in operation, and from the size of this jet the flow per boiler hour is calculated and reduced to B.t.u.

12. Steam charged to forced-draft engines and turbines: From tests made on this equipment the water rates under average operating conditions have been determined, and from these water rates and the time that engines and turbines are in operation, the total

steam flow is calculated and reduced to B.t.u. Since the exhaust steam is returned to the exhaust-steam header after passing through an oil separator, 20 per cent of the heat content is charged against forced draft and the remainder is considered as going to the exhaust and is taken care of in the steam distribution.

13. Steam used by soot blowers: From the number and size of nozzles and the time the soot blowers are operated the steam flow has been calculated and reduced to flow per boiler hour. This flow is reduced to B.t.u. by using the average B.t.u. per pound.

14. The net amount of steam produced is determined by deducting from item 9 the sum of items 10, 11, 12 and 13.

15. Steam used by boiler-feed pumps: From tests made on the pumps a figure of B.t.u. per pound of feed water pumped has been determined. From the venturimeter charts for the boiler feed the amount of water handled by the reciprocating, turbine- and motor-driven pumps is segregated so that the proper constant can be applied to each. As with the forced-draft drives 20 per cent is charged to the pumps, the remainder going to the exhaust.

TABLE II—DISTRIBUTION OF AVAILABLE STEAM

		April, 1921	
		Exhaust Million B.t.u.	Equipment Million B.t.u.
A	Air compressors	2,062	515
B	Turbo-generators	5,232	1,308
C	Hydraulic-elevator pumps	3,170	793
D	House pumps	1,324	331
E	Fire pumps	816	204
F	Hot-water plumbing supply	.....	3,379
G	Hot-water supply, heating	.....	13,610
H	Steam for building heating	.....	40,173
J	Steam for miscellaneous use	.....	9,713
K	Live steam to header for items *F* and *G*	384	.....
L	Exhaust from boiler-room auxiliaries	4,001	.....
	Total	16,989	70,026

16. Steam used by the skip-hoist engine: This engine is used to operate a hoist which raises the coal to the distributing conveyors over the bunkers. The amount of steam consumed is calculated from a test figure of B.t.u. per gross ton of coal handled and the gross tons of coal received during the month. Twenty per cent of the heat content is charged to the engine and the remainder to exhaust.

In the winter months steam is used in the coils around the coal and also to warm the skip-hoist engine room. The amount used for these purposes is calculated from the square feet of radiation and reduced to B.t.u. The resulting figure is added to the amount charged to the skip-hoist engine.

17. Loss in the feed-water heater: This is estimated at a certain percentage of the total heat in the feed water above the temperature of the makeup. Normally, the temperature of the feed water ranges between 200 and 206 deg. F. When there is an excessive vent loss, it is noticed that the feed-water temperature drops. From the weight of feed water and the difference in temperature between the low and the normal temperature an estimate can be made of the vent loss.

18. Line losses by radiation: This is taken as 0.1 per cent of the heat in the net steam produced.

19. Net steam available for distribution is calculated by deducting from item 14, the sum of items 15, 16, 17 and 18. This is the total to be distributed on the distribution of steam, Table II.

20. Steam equivalent of the electric power for operating stokers, is determined from the B.t.u. necessary to produce a kilowatt-hour, the test figure of kilowatt-hours consumed per stoker hour and the number of hours the stokers are operated.

21. Steam equivalent of the electric power for operating the coal conveyors is calculated from the B.t.u. per kilowatt-hour and a test figure of kilowatt-hours per gross ton handled.

22. Steam equivalent of the electric power of operating boiler-feed pumps is calculated from the B.t.u. per kilowatt-hour and the kilowatt-hours taken by the pumps as measured by a watt-hour meter.

23. The boiler efficiency is determined from the B.t.u. in the total equivalent evaporation and the total heat in the coal as fired (item 9 divided by item 2 in column D). At the plant in question boiler efficiency depends on the number of hand-fired boilers in service and the grade of coal used. During the past summer anthracite yard screenings were burned with success but with a loss in boiler efficiency as compared with burning No. 2 buckwheat, which is used during the winter months. A mixture of No. 2 buckwheat and screenings has also been burned with success. When screenings are burned, it has been found that the ashpit loss increases, resulting in a lower efficiency.

24. Over-all plant efficiency is calculated by deducting from the total B.t.u. in the steam available for distribution, the B.t.u. in the steam equivalents of the electrically driven apparatus, and dividing the result by total B.t.u. in the coal as fired, item 2.

The charts from the boiler steam-flow meters are gone over daily and any defects in operating practice are corrected. These defects will naturally lower the boiler efficiency for the month, and when the charts do not offer a solution the next place to look is at the coal or in the ashpit.

In the winter months live steam is supplied to the exhaust header to make up for the deficiency in exhaust steam and the amount of this live steam is measured by means of a steam-flow meter. From the exhaust header steam is taken by the heaters for the hot-water heating system, the heater for the plumbing hot-water service and by the feed-water heater. The amount supplied to the first two sets of heaters is calculated, and the difference between this amount and the total supplied goes to the feed-water heater. Comparison of this item with item 5b (all other condensate) after an estimate has been made for the drip returns, will give an indication of the actual vent loss in the feed-water heater.

Distribution of Available Energy

Referring to Table II, distribution of available steam, each item will be taken up in turn. In items A to E, 20 per cent of the energy is charged to the equipment and 80 per cent to exhaust.

A. Air compressors: For average load conditions a test figure has been determined giving the B.t.u. per stroke required for each compressor. The number of strokes is obtained from a stroke counter from which the total B.t.u. is calculated.

B. Turbo-generators: From the water-rate curves of the turbines the pounds of steam per kilowatt-hour are determined and reduced to B.t.u.

C, D and E. Elevator, house and fire pumps: The amount chargeable to each is calculated from a test figure of B.t.u. per stroke and the number of strokes as taken from a stroke counter.

F. Hot-water plumbing supply: This is composed of two items, the heat abstracted from the water circulated and the heat added to the makeup water. The circula-

tion is measured by means of a turbine-type water meter, while thermometers record the temperature of the supply and return. The circulation in cubic feet is changed to pounds by multiplying by the weight per cubic foot corresponding to the temperature of the supply, and the pounds multiplied by the drop in temperature. The difference between the supply and return temperatures gives the B.t.u. abstracted from the water. From the cubic feet of makeup water as metered by a turbine-type water meter, reduced to pounds by multiplying by the pounds per cubic foot corresponding to the temperature of the makeup, and the difference in temperature between the supply and the makeup, the amount of heat added to the makeup is calculated. Adding together, the figures just obtained and dividing by the assumed efficiency of distribution (98 per cent) from the header to the point of measurement, gives item F.

G. Hot-water heating: There are three different systems in this plant, known as the low-, intermediate- and high-pressure systems, each system designed to heat a certain number of stories. A venturi meter registers the flow in gallons and a differential recording thermometer records the drop in temperature. Gallons are reduced to pounds, and the product of the pounds and the drop gives the total heat abstracted from the circulating hot water. The makeup is also measured in cubic feet and the temperatures of supply and return are recorded. The cubic feet are reduced to pounds, and the product of the weight of makeup and the difference in temperature between the supply and makeup gives the heat added to the makeup. The sum of the two results just obtained divided by the assumed efficiency of distribution from header to heater gives item G.

H and J. The steam supplied to buildings and for miscellaneous purposes is measured by means of steam-flow meters. Pounds are changed to B.t.u. by multiplying by the B.t.u. per pound above the temperature of makeup water corresponding to the pressure of the steam metered.

The sum of items A to J should equal the total steam available for distribution. This rarely occurs, as there is a slight discrepancy due to meter errors and the application of test constants. Adjustments are made in items H and J to take care of the discrepancy.

K. This is the live steam required in addition to the exhaust obtained under items A, B, C, D, E and L, to furnish the heat for items F and G.

L. This is the heat in the exhaust from the forced-draft engines and turbines, the boiler-feed pumps and the skip-hoist engine.

Conclusions

The heat balance and distribution of steam were primarily developed for the purpose of obtaining a basis for allocating the expense of operating the plant. An interesting accounting system has been developed so that the production cost of each service is determined separately. A portion of all services supplied is sold to tenants, the rates charged by the public-utility companies being the ruling rates for electric power and steam with an adaptation in the case of hot water, and production cost plus a slight profit applying to the other service.

Some of the methods used to calculate the heat balance and the distribution of steam are open to criticism, but the results obtained serve the purposes for which the calculations are made.

Relation Between Width of Belt and Power Transmitted

The results of research being carried out at Cornell University by the Leather Belting Exchange Foundation have been summarized from time to time in *Power*. Report No. 108, recently issued by R. F. Jones, Research Engineer of the Foundation, gives the results of and conclusions from experiments undertaken to find the relation between the width of a belt and the power that can be transmitted at various percentages of slip.

The experiments were carried out with a belt 6.05 in. wide, weighing 16.5 oz. per sq.ft., and having a thickness of 0.205 in. These measurements were taken before testing. The general method of testing was to run the belt in thoroughly until a condition of constant capacity was reached, after which the width was reduced 1 in. at a time and a test made at each width to determine the relative capacity. In each case a slow-running tension of 72 lb. per inch of width of belt was used.

The belt used was endless and ran over 24-in. cast-iron pulleys with a belt speed between 3,100 and 3,200 ft. per min. For any given slip (up to 1½ per cent) the power transmitted was found to be in proportion to the width of the belt. At higher slips, ranging up to 3 per cent, there was a slight falling off in the rate at which the power increased with the width of the belt after the latter had passed 4 inches.

Further experiments will be required to find out whether this falling off was due to experimental errors. The matter is not of great practical importance because belts are not ordinarily designed to run with a slip greater than 1½ per cent. Within this range the power transmitted is closely proportional to the width, as might be expected.

It should be noted that this relation has been proved only for the particular belt tested and for the conditions under which it was tested (given tension per inch of width, belt speed, size of pulley, etc.). However, it is reasonable to expect that the power transmitted at a given slip would, in all cases, be practically in proportion to the width.

Starting of Ammonia Compressor

The ammonia compressor should never be started until the engineer has first made sure that the discharge-valve line is open. He should not forget to turn the water on the condenser coils before or immediately after starting the compressor.

The head or condenser pressure should be watched upon starting the compressor, as it will warn the engineer if things are wrong.

The suction valve should not be opened hurriedly upon starting, especially if the machine has been shut down for a long time without having the expansion coils and suction lines pumped out. The operator should keep his hand on the discharge valve if possible. If it gets cold suddenly and if the discharge valve and compressor work noisily, liquid is likely coming back with the gas, and the suction valve should be closed off until the machine begins to warm up. If there is a slamming in the cylinder, the suction valve should be closed until the noise stops. This slam is in most cases due to the presence of some liquid ammonia coming in from the suction line.

New Condensers for an Old Turbine

Baltimore Plant Makes Unique Application of Low-Level, Multi-Jet Condensers

SOME time ago it became necessary to replace a pair of wornout low-level jet condensers attached to a 7,500-kw. turbine in the Westport Station of the Consolidated Gas, Electric Light and Power Co., of Baltimore, Md. Although the character of the water supply available at the Westport plant would seem to make the installation of a surface-type condenser the logical step, a careful analysis of the problem by A. L. Penniman, superintendent of the steam stations of the company, led to the selection of the jet type of condenser, principally because of its low first cost. Cost, rather than economy of operation, became the deciding factor, because the turbine had already seen ten years of service and would probably wear out or become obsolete in a relatively short time, and also because the load factor was low. The plant, operating in connection with water-power stations, has newer and larger units to carry the base load, so this unit practically becomes a stand-by. In addition the particular condenser chosen could be installed with a minimum of alterations.

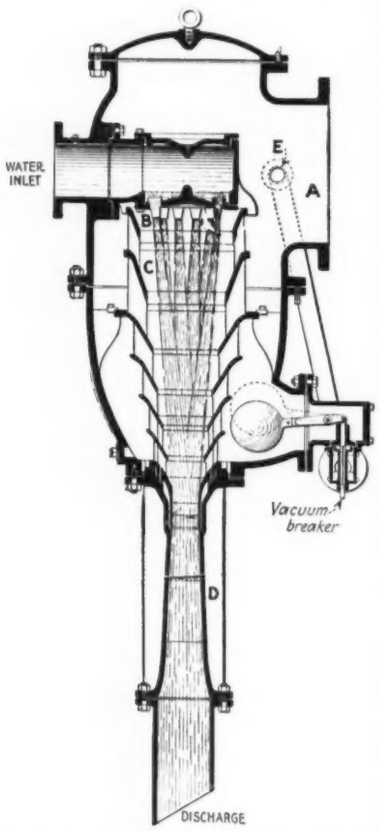


FIG. 1. CROSS-SECTION OF LOW-LEVEL, MULTI-JET CONDENSER

The condensing plant consisted of Ingersoll-Rand, twin low-level, multi-jet condensers and twin injection pumps. Fig. 1 shows a section of this type of condenser. Steam enters the condensing head through the flanged opened *A* at the right. The condensing water, entering at the opposite side, is projected downward from the ring of nozzles *B* through the combining tube *C*. The steam is condensed by the jets and the resulting condensate, together with the air and non-condensible vapors, is entrained and carried by their momentum out through the venturi tube *D* to the atmospheric discharge below.

The customary vacuum pump and the more-or-less inaccessible removal pump are not needed. The only condenser auxiliary is the centrifugal pump, which supplies cold injection water to the ring of nozzles at 18 or 20 ft. head. This injection pump, unlike the removal pump which withdraws hot water from a vacuum, is not handicapped by a 28 or 30-ft. suction lift. It is only necessary to prime and start the injection pumps in order to put the turbine under vacuum.

The general arrangement of the condensing plant is shown in plan and elevation in Figs. 2 and 3. No expansion joint is used; the exhaust connection is bolted directly to the turbine exhaust, and the weight of the condensers is taken by columns under brackets cast on the condenser heads. These brackets normally are cast on the lower half of the condenser body. In this case, at the request of Mr. Penniman, they were moved up close to the top of the condenser head, thus reducing the vertical distance between the turbine base and the top of the supporting columns to but a few feet. Changes in vertical distances due to variations in temperature thus become so small that they are absorbed by the exhaust connections and the columns. The latter are light *I*-beams located, as may be seen in Fig. 2, about 13 ft. from exhaust the center line.

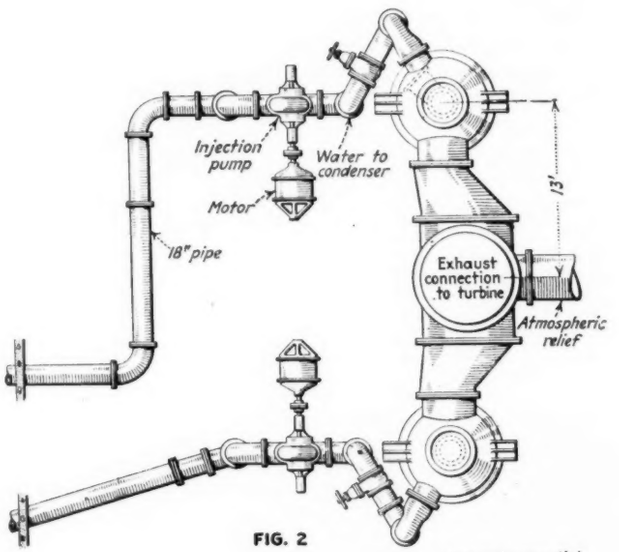


FIG. 2

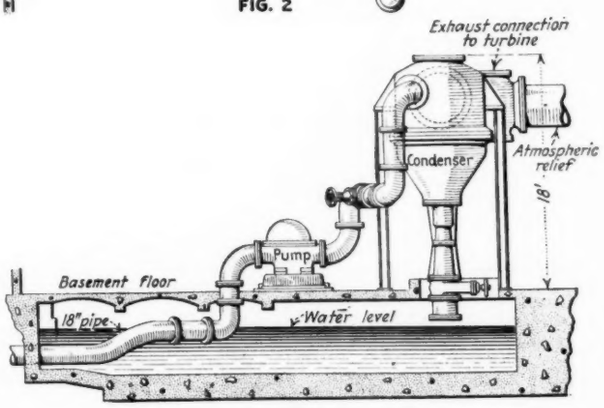


FIG. 3

FIGS. 2 AND 3. PAIR OF MULTI-JET CONDENSERS WITH PUMPS AND CONNECTIONS

Fig. 2—Plan. Fig. 3—Elevation.

Had it not been a case of using the exhaust connection already in place, this elongated construction would have been replaced by the standard twin multi-jet construction. This combines the twin condensing heads into a single chamber with a flanged steam nozzle at the top and twin discharge legs suspended below, the whole condenser being hung directly from the turbine exhaust.

Fig. 4 is a photograph of one of the condensing units, showing the Cameron injection pump and piping. One of the vacuum breakers for this condenser may be seen mounted on the left side of the body; these contain the only moving parts outside of the pump. Two of these breakers are supplied with each condenser and prevent flooding by admitting air should the condenser head begin to fill. Such an occurrence is highly improbable because the condenser discharge is above the water level in the discharge canal, as shown in Fig. 3.

Another method of obtaining the same result would be to have the portion of the hotwell immediately surrounding the condenser discharge separated from the rest by a weir. In this way only a limited quantity of water could be drawn up into the condenser.

Full-load steam and water quantities are included with other data in the following table. It should be noted that, while the generator rating is only 7,500 kw., the quantity of steam condensed is nearer the requirement for a modern 10,000-kw. unit.

Steam quantity, for each condenser, lb. per hr.	56,000
Injection water to each condenser, gal. per min.	10,000
Vacuum (with water at 79 deg. F.), inches	28.0
Pump speed, r.p.m.	720
Pump motor, horsepower (40 deg. F. rating)	150
Pump horsepower per kilowatt of main unit	0.04
Diameter of condenser heads at flange, ft.	8
Diameter of steam nozzle, ft.	4
Diameter of water nozzle, in.	22
Elevation of center line of steam nozzle above hot well, ft.	15.5
Approximate shipping weight each, lb.	21,000

As may be seen from the illustrations valves are provided in the injection lines and in each condenser discharge. Under conditions of cold water or light load, one of the condensers can be cut out of service by closing its injection and discharge valves, thus reducing the auxiliary power consumption by 50 per cent. From the performance curves, Fig. 5, it is evident that this feature permits carrying full load on one condenser at better than 28 in. vacuum for about four months during the winter.

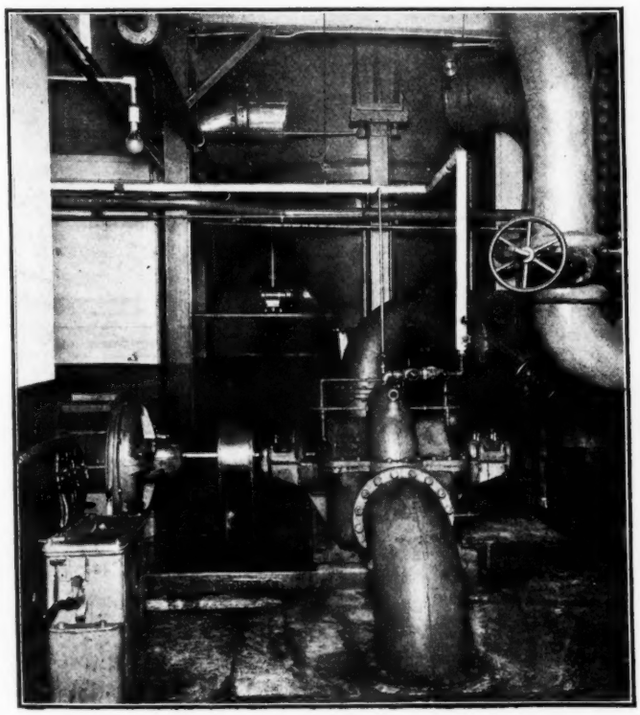


FIG. 4—SHOWING ONE CONDENSER WITH ITS MOTOR-DRIVEN PUMP

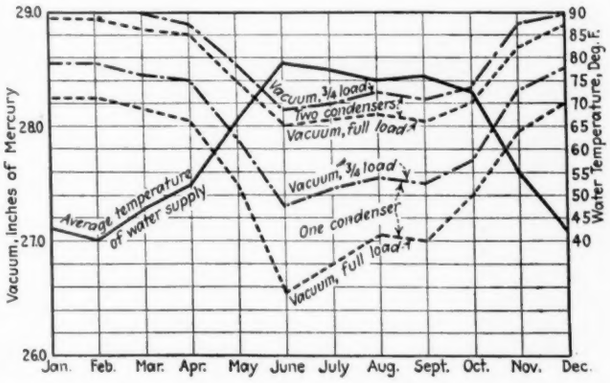


FIG. 5. GUARANTEED PERFORMANCE FOR TWIN UNIT AND FOR SINGLE CONDENSER, AT FULL LOAD AND AT THREE-QUARTER LOAD

The solid curve in Fig. 5 shows the average water temperature during each month of the year. The corresponding vacua at full and also at three-quarters load, with either one or both condensers operating, are shown by the four dotted curves. The guarantee is not, of course, based on the month of the year, but on the actual water temperature. For any given temperature this guarantee is easily determined by running horizontally from the temperature scale to the temperature curve and thence up or down to the dotted curve corresponding to the load carried and the number of condensers used.

Shortly after installation the unit was tested by the company under practically full-load conditions, with the following results:

Average load on unit, kw.	7,450
Steam condensed (by flow meter), lb. per hr.	111,246
Injection water (from pressure at nozzles), gal. per min.	20,585
Injection-water temperature, deg. F.	70.9
Ratio pounds of injection water per pound of steam	92.4
Vacuum in condenser (referred to a 30-in. barometer), in.	28.33
Vacuum guaranteed under existing load and water conditions, in.	28.25

The smoke problem is nearly as old as civilization itself, for coal was being mined in Great Britain about the Tenth Century. In 1306 Edward I issued a proclamation prohibiting the burning of coal in the City of London, because of the "sulferous smoke and savour of the firing." In 1648 the people of London petitioned Parliament to prevent the importation of coal into the city. Beginning about 1750, when steam was applied to industrial purposes, coal began to be used more extensively, and consequently smoke became more abundant. In 1819 the Government appointed a House of Commons committee to inquire into the matter, and another committee was appointed in 1843. Two reports were issued in 1843 and 1845, but no legislation was passed. Other investigations were made in 1846, 1854 and 1866, and finally, in 1875, the Public Health Act was passed, dealing with the question of smoke.—*Bureau of Mines Investigation Report No. 2323.*

Electromagnets and How They Operate

How an Electric Current Flowing Through a Coil of Wire Causes the Coil to Become a Magnet

By B. A. BRIGGS

WHEN an electric current flows through a coil of wire whose turns are all wound in the same direction, the coil takes on all the properties of a magnet. One end of the coil will have a North pole and the other end a South pole, and if suspended by a string so that its poles are in a horizontal plane, as in Fig. 1, it will point north and south, if free to

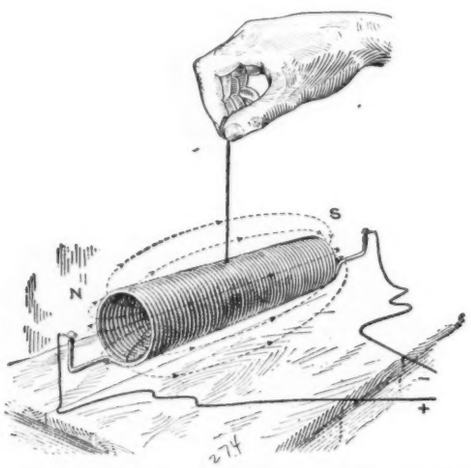
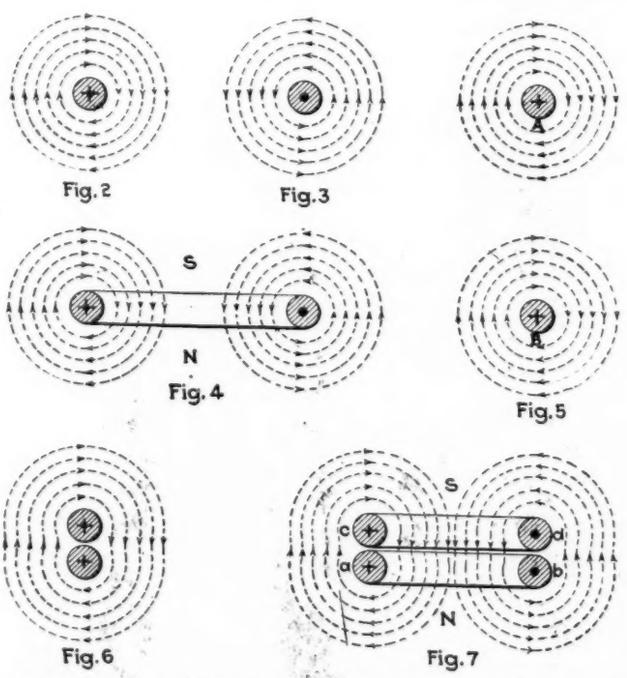


FIG. 1. AN ELECTRIC CURRENT FLOWING THROUGH A COIL OF WIRE GIVES IT THE PROPERTIES OF A MAGNET

move, just as a compass needle will. The coil will also attract pieces of iron the same as a permanent magnet.

These properties are imparted to the coil owing to the fact that an electric current, when flowing in a conductor, sets up a magnetic field. This magnetic field takes the form of circles which move in a definite direction about the conductor, depending on the direction of the current. If the current is flowing away from the observer, as indicated in Fig. 2, the magnetic field will have a clockwise direction about the conductor. When the current is flowing toward the observer, as indicated by the dot in the center of the circle, Fig. 3, the magnetic field will have a counterclockwise direction, as shown by the arrowheads. This is just what makes possible an electromagnet, which is what a coil of wire with an electric current flowing through it is called. It will be noted that the lines of force between the conductors, Figs. 2 and 3, are in the same direction. Therefore, if the current is caused to flow through a loop, as in Fig. 4, which is one-half turn of a coil, the lines of force will enter the loop at the top and leave at the bottom. This is identical to what we have in a permanent bar magnet, and the end from which the lines of force leave is called the North (N) pole and the end where the lines enter is called the South (S) pole. What is shown in Fig. 4 is the equivalent of an electromagnet having one turn.

If we have two conductors *A* and *B*, both carrying current in the same direction, as in Fig. 5, it will be seen that between the two conductors the lines of force have opposite directions. On the bottom side of *A* the direction of the lines of force is to the left, while on the top side of *B* it is to the right. It is evident from this that if the two conductors are brought near to each other, the two magnetic fields between them will oppose each other. Fig. 6 shows the results of bringing the two conductors together. The lines of force between the conductors oppose each other, therefore cannot take this path, so they have combined to form one field which surrounds the two as shown. Referring to Fig. 5 again, it will be found that the lines on the left-hand side of each conductor have upward direction while those on the right hand have downward direction, and this compares with the condition in Fig. 6. Comparing Fig. 6 with Fig. 2, it is seen that they are practically alike, with the exception that in the magnetic field, Fig. 6, the vertical axis is longer than the horizontal axis owing to the presence of the two conductors. This is just what happens in a coil of two or more turns, which will be found by an inspection of Fig. 7. In this figure there are two turns instead of one, as in Fig. 4, but since the direction of magnetic fields oppose between the turns, the two fields combine to form but one as indicated. In an actual coil we would



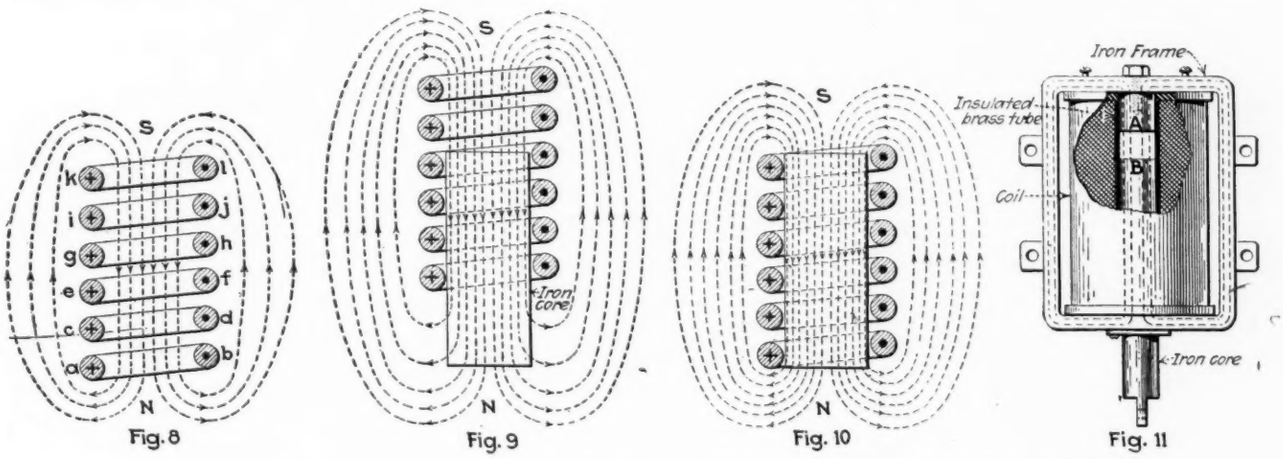
FIGS. 2 TO 7. MAGNETIC FIELD SET UP ABOUT CONDUCTORS BY AN ELECTRIC CURRENT

have current flowing in at, say, *a*, out at *b* into *c* and out at *d*, but for clearness only one-half of each turn is shown.

What has been shown for two turns in a coil is true for any number of turns. In Fig. 8 a section through

a coil of 6 turns is shown, and we might consider the current entering at *a* coming out at *b* and entering the next turn at *c* and so on until the current came out at *l*. The magnetic fields that this current tends to set up about each conductor will combine to give one field as indicated. By comparing Figs. 7 and 8, it will be seen that Fig. 8 is only an enlargement of the principle shown in Fig. 7. If an iron core is placed in the coil, Fig. 8, in some such position as indicated in Fig. 9, two things would happen: (1) The magnetic field would become stronger due to the iron core being a better conductor of magnetism than air, and (2) the coil would exert a pull on the iron core. The latter is important since it is the principle involved in a great many electrical devices. One of the applications is to operate motor controllers automatically. If the core were free to move and the magnetic pull sufficient to lift it, the result would be that indicated in Fig. 10, where the core is shown drawn up into the coil. The simplest explanation for this is that lines of force are like elastic bands in tension, in that they tend to become shorter. In Fig. 9, if the lines that represent iron, the longer the air gap between *A* and *B* the smaller will be the number of lines of force in the air gap and the smaller the pull tending to lift *B*. When core *B* moves toward *A*, the air gap is reduced, and as this reduces the resistance of the magnetic circuit, the number of lines of force will be increased, which will result in an increased pull on *B*. From this it is evident that the nearer core *B* gets to *A* the stronger the pull will be between the two, and the farther apart the cores are the smaller the pull. If the distance between *A* and *B* is made too great, the magnetic flux will be reduced to values where the magnetic pull will not be sufficient to lift the movable core. The variation in pull between the movable and the stationary cores due to the distance between them, is utilized as a means of adjusting the pull of electromagnets on motor controllers.

Since the magnetic pull is many times greater when the cores are together than when they are apart, there need not be as much current flowing through the coil after the movable core is lifted. For this reason it is general practice to arrange the connections to the coil so that



FIGS. 8 TO 11. SHOW AN ELECTROMAGNET IN DIFFERENT FORMS OF DEVELOPMENT

the magnetic field were elastic bands, they would tend to pull the core to the position shown in Fig. 10, where they would have reached their minimum length, and pass in one end of the coil and out at the other.

Electromagnets, as used on motor controllers, are usually ironclad to make them more efficient. Iron being many times a better conductor than air, the nearer that the path for the line of force can be made completely of iron, the stronger the magnetic field will be for a given current and size of coil. In one type of electromagnet the core and frame are arranged as indicated in Fig. 11. The coil is wound on a fiber bobbin, and when assembled in the frame a brass tube is placed inside, which acts to hold the coil in place and also to guide the core in it. The core is usually made in two parts, a short piece which is held to the top of the frame by a bolt, and a longer part which is the movable element and is attached to whatever part of the controller is to be operated. When current flows in the coil, lines of force are set up in the core and frame, as indicated. This creates a strong pull between *A* and *B*. The lines of force in the air gap between the stationary part of the core *A* and the movable part *B* acting as elastic bands in tension tend to lift *B*.

As air is a much poorer conductor of magnetism than during the time of travel of the core the coil is connected directly across the line. When the cores come together, a high resistance is cut in series with the coil, which cuts the current down to about 25 per cent of what it was when the core was being pulled up. This scheme greatly reduces the amount of power required to hold the magnet closed, and at the same time allows the use of a much smaller magnet to do a given work. These features will be discussed further in future articles on controllers.

Two years without a single lost-time accident is the record that A. L. Brown, power superintendent of the du Pont de Nemours Dye Works, at Wilmington, Del., writes about with pardonable pride in the March issue of the *National Safety News*. Losses from accidents had previously been serious. Weekly meetings were held and safety enthusiastically discussed. Each man was asked to write a note to the power superintendent, telling what he thought to be the most unsafe operation in the department. As a result, dangerous conditions were brought to light and suitable safety devices installed. "The chief single factor that has made this record possible," says Mr. Brown, "is the safety spirit of the entire department."

Why Complete Compression Is Not Economical

A DISCUSSION of the compression line on the indicator diagram, showing that compression to the initial pressure is not efficient. Description of a graphical method of finding proper compression for any given cutoff.

WHEN the piston has completed its return or back-pressure stroke, the exhaust valve is closed and the steam valve opens, admitting boiler steam into the space between the cylinder head and the piston. If the piston came up against the cylinder head and if there were no port spaces, the volume to be filled with steam would be that part of the stroke volume from the beginning of the stroke to cutoff, *A* to *B* in Fig. 1. Complete expansion of this volume of steam would carry the pressure to *D*, and the back-pressure line *D* would continue to the extreme end of the return stroke *Q*.

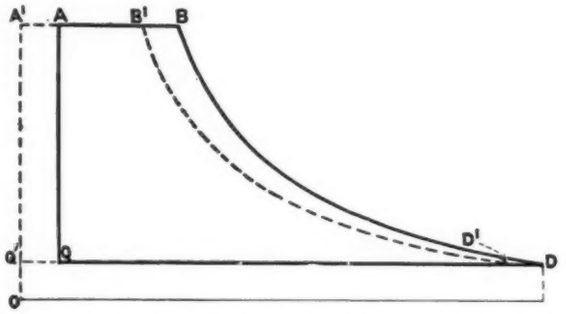


FIG. 1. COMPLETE EXPANSION IN ENGINE WITHOUT CLEARANCE

In engine construction it is necessary to have some clearance between the piston and the cylinder head, and in addition there is the port volume to be filled with the boiler steam. The result is that the steam on entering the cylinder, first fills these clearance spaces, the volume of which is indicated by the length *A'A* in Fig. 1. If the same volume of steam is admitted into the cylinder now as for the case with no clearance, the cutoff would be at *B'*, the volume *BB'* being equal to *AA'*, the clearance volume. The expansion now, if complete, is from *B'* to *D'*. The area *BDD'B*=*A'AQQ'* is lost by the presence of clearance in the engine.

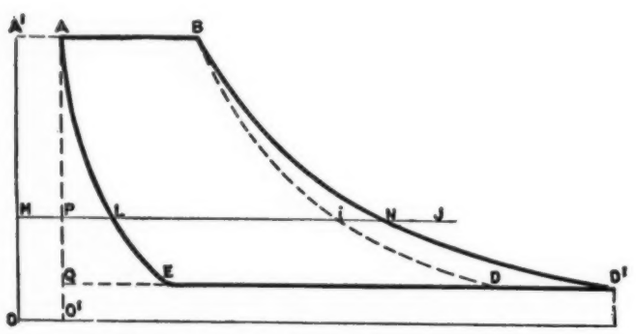


FIG. 2. COMPRESSION AND EXPANSION

If, now, the exhaust valve is closed at some point *E*, Fig. 2, before the end of the stroke, the piston, by compressing the steam into the clearance space, would raise the pressure of this steam, trapped in the cylinder by the closing of the exhaust valve, until at the end of the stroke it would equal the boiler pressure at *A*. The compression would follow the curve *EA*. Since the clearance would now be filled with steam of boiler pressure, the steam fed to the cylinder from the boiler would then be *AB*, in Fig. 2 and equal to *AB* in Fig. 1, for the engine without clearance. The expansion without clearance would be *BD*, while the expansion with clearance and with the compression up to the point *A* would be *BD'*, for we are dealing with a volume proportional to *AB* in the first instance, and with this volume, plus the clearance volume, or *A'B* in the engine with clearance.

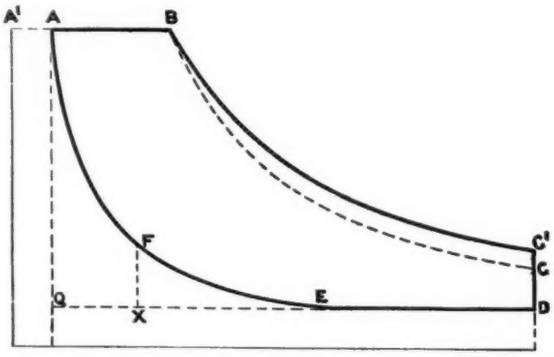


FIG. 3. LOSS WITH INCOMPLETE EXPANSION IN ENGINE HAVING COMPLETE COMPRESSION

If steam were a perfect gas and if no condensation or change of condition occurred, the loss of clearance pointed out would be eliminated by the compression of the gas to the initial pressure at *A*. In the diagram, Fig. 2, let us assume that at the end of the stroke indicated at *AA'* a diaphragm is stretched across the

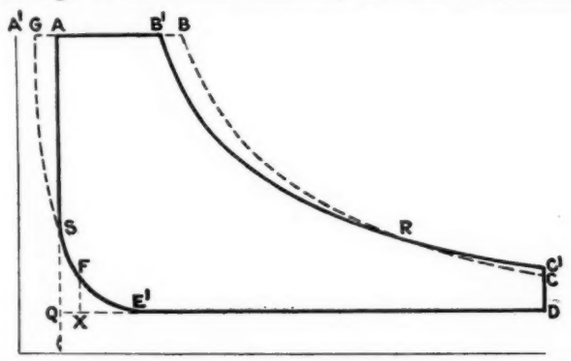


FIG. 4. DIAGRAM SHOWING INCOMPLETE COMPRESSION

cylinder. Between the cylinder head and the diaphragm is placed the clearance steam AA' at a pressure equal to $A'O$. Now, as the piston starts on its working stroke, the live steam flows in at boiler pressure up to the point of cutoff B. The diaphragm has on one side the clearance steam at the pressure $A'O$, while on the piston

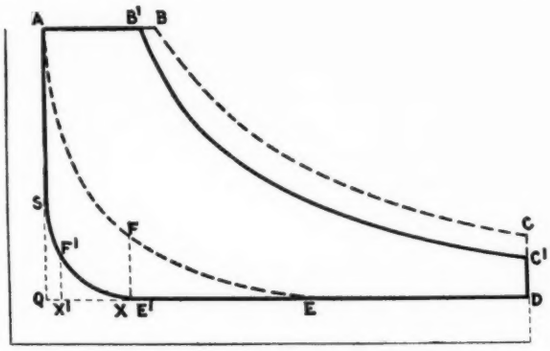


FIG. 5. COMPARISON OF COMPLETE AND INCOMPLETE COMPRESSION

side the cylinder feed at the same pressure is acting against the diaphragm. At B the expansion begins, and as the pressure drops the clearance steam also expands, forcing the diaphragm toward the center of the cylinder. The steam supplied to the cylinder, indicated by AB, expands along BD, the volumes being measured from AO', while the clearance steam expands along the line AE measured from the zero clearance line $A'O$. The total volume of the cylinder feed and clearance feed at any pressure indicated by the line MJ is the sum of the volume of clearance feed ML plus the volume of the cylinder feed PI. The total expansion volume is then MN, where IN is equal to PL.

The combined expansion curve, as made by the indicator pencil on the indicator diagram, is BD'. On the back-pressure stroke the cylinder steam is expelled by the time the piston reaches E. Further movement of the piston causes the clearance steam to be compressed along the lines EA, giving the diagram $ABD'EA$. With no compression and no clearance the diagram would be $ABDQA$. We have lost the area EAQ, but have gained the area BDD'. As pointed out, IN is equal to PL, and this relation exists during the entire process of expansion, so that we can say that the two areas are equal, and that by complete compression we have secured the same diagram area as obtained in Figs. 1 and 2 with no compression and no clearance. This, however, applies only in case the expansion is complete to the back-pressure line. Furthermore, it is true only for a perfect gas without condensation.

Effect of Incomplete Expansion

In the steam engine expansion is rarely complete, and an entirely new set of conditions exist. Considering the action of a perfect gas, and on the assumption first that the engine is without clearance, the diagram $ABCDQA$, Fig. 3, is obtained with incomplete expansion. The weight of steam supplied to the engine is proportional to the length AB. Next consider the engine with clearance AA' and complete compression from E to A. Presuming, as before, that the clearance steam is separated from the cylinder steam by a diaphragm, it is evident that the clearance steam volume and pressure during expansion and compression is along EA, while the volume and pressure of the cylinder steam AB is along the curve BC. The total volume during expansion follows the curve BC'. Since the end of the stroke is fixed by the stroke volume, the cylinder steam is discharged along $C'D$ and DE and the expansion of the clearance steam from F to E does no work on the piston, which is at the end of its stroke, but merely expands as the cylinder pressure drop from C' to D. The work of compressing the steam from E to F is the area EFX and, as explained, is not returned on the re-expansion stroke. An engine with incomplete expansion, with clearance and compression to the initial pressure does not return all of the work of compression and is then less efficient than the engine with no clearance, owing to the loss of the area EFX, Fig. 3.

Loss with Incomplete Compression

Assume that the compression is not complete to the initial pressure at A, but that the exhaust valve closes at some point E', Fig. 4. The compression of the compression steam will be along the curve $E'S$, giving a pressure of S when the piston reaches the end of its stroke. When the steam valve admits live steam into the clearance space, this high-pressure steam will continue to compress the clearance steam along $E'S$ to G. The clearance volume to be filled with live steam is then AG in Fig. 4. Disregarding for the moment the compression line $E'S$, the area loss due to the existence of the clearance which must be filled with live steam is the area BRB', and the gain is the area CRC', where BC is the expansion line of the same amount of steam fed to the cylinder along AB with no clearance, and the curve $B'C'$ is the expansion of the steam fed into

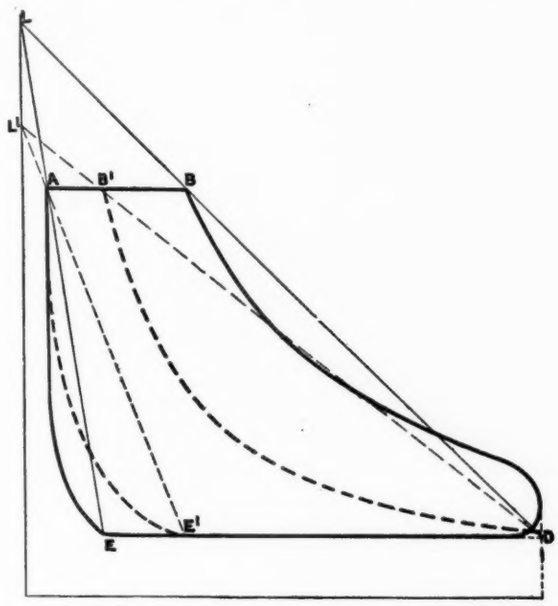


FIG. 6. GRAPHICAL METHOD OF FINDING PROPER COMPRESSION

the cylinder from G to B', being equal to AB. In addition, the re-expansion of the compression steam along GE' exerts a force upon the piston only to F, the pressure at F corresponding to the pressure C' at the end of the stroke. The work of the compression from E' to F, or the area $E'FX$ is not returned during expansion. The net loss from compression and clearance is then the areas $BRB' + E'FX - CRC$.

A comparison of complete and incomplete compression in Fig. 5 gives for complete compression the diagram *ABCDEA* and for incomplete compression the area *AB'C'DE'S*. Complete compression adds the area *BCC'B'*, but the area *EASE'* plus the difference between *EFX* and *E'F'X'* is lost. Clearly, at some compression other than complete the highest efficiency is obtained. This is emphasized by a consideration of the heat loss to the cylinder walls.

The steam during the back-pressure stroke is dry saturated and in some instances is superheated a few

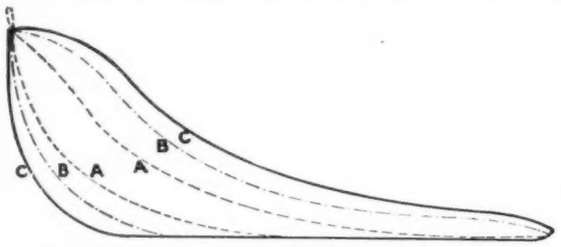


FIG. 7. SINGLE-VALVE ENGINE DIAGRAMS WITH VARIABLE LOAD

degrees. If the steam trapped in the cylinder by the closing of the exhaust valve did not absorb or give off heat to the cylinder walls, the compression would be adiabatic, as would likewise the expansion line *AE'*. Since the walls do absorb part of the heat of the steam both during compression and re-expansion, all the work of the compression is not returned during re-expansion period even with complete expansion and compression. It has been demonstrated mathematically that the proper compression may be determined in an engine with a given percentage of clearance for any chosen cutoff. This method, shown graphically, is illustrated in Fig. 6. If on the diagram with cutoff at *B* the line *LD* be passed through the points *B* and *D* at the end of the stroke, giving the point *L* on the clearance line, then a line *LE* drawn through the point *A* on the diagram will give the point *E* at which compression should begin were the cutoff at *B*. An earlier cutoff at *B'* gives an earlier compression at *E'*. The method indicates that in an engine having a variable cutoff the compression should vary inversely with the load; namely, the earlier the cutoff the greater the compression ratio. This is the way the valve events take place in a shaft-governed automatic engine as shown in Fig. 7, where *AA* are the expansion and compression lines at low load, *BB* like curves at half load and *CC* at full load. The principle upon which the foregoing graphical method is based may be explained in the form of a rule.

Finding Correct Compression

Rule 1. The clearance volume should bear the same ratio to the volume at the commencement of compression that the volume at cutoff bears to the volume at the end of the stroke.

Considered from the mechanical side alone, some compression is necessary to absorb the momentum of the piston, crosshead and connecting rod. This momentum acts to oppose the efforts of the steam entering the clearance space to reverse the piston. When the crank approaches head dead center, as outlined in Fig. 8, the piston and connecting rod are in tension, being pushed to the left by the residual steam pressure in the crank end of the cylinder and the momentum of the parts. If the steam valve has no lead and opens at dead center, at the instant the piston reaches the end of the stroke the live steam rushes into the cylinder, strikes the head-end face of the piston, and the piston and connecting rods are thrown from a state of tension to that of compression. The lost motion in the wristpin and piston-pin bearings causes the parts to pound violently. If the live steam could be admitted gradually and so enable the pressure in the head end to build up slowly, the parts would come to rest without shock. This is not feasible mechanically since the full steam pressure would not be exerted immediately on the beginning of the stroke due to the wiredrawing of the steam through the valve, and the power secured from each pound of steam would be decreased.

To enable the piston and reciprocating parts to come to rest without shock, the exhaust valve must be closed somewhat before the end of the stroke and the steam trapped in the cylinder compressed by the further movement of the piston, providing a gradually increasing pressure in front of the advancing piston, and the reciprocating parts are thereby brought to rest quietly and without shock.

Engine with Fixed Compression

In engines with a fixed point of exhaust-valve closing, such as the Corliss, the compression should be as low as possible without pounding of the bearings. The curve should merge into the admission line as in Fig. 11b.

The back pressure has an important influence on the closing point of the exhaust valve. In Fig. 9 are several diagrams with different back pressures, all of which show the exhaust valve closing at *E*. This gives the diagram with the low back pressure a low final compression. It is necessary to make the compression start earlier if the back pressure is reduced on an engine.

High-speed single-valve engines often give a diagram on lod loads with a loop at the end of compression, as

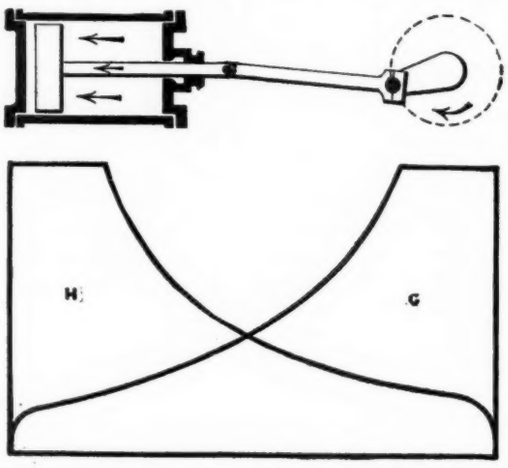


FIG. 8. EFFECT OF PISTON AND ROD INERTIA

shown at Fig. 10 *b*. Early compression which accompanies short cutoff in this type of engine, will bring the final compression above the steam-chest pressure. Upon the uncovering of the steam port, which occurs a little before dead center, the compression promptly falls to that of the steam chest giving the loop shown. Unaflow engines operating condensing may at times show the

same characteristic loop when the vacuum is lost and the exhaust line is extra long. The long line will hold up the exhaust pressure, and as the auxiliary exhaust valve closes early, the compression may rise above boiler pressure.

Compression lines of the type shown at *c* are frequently found. This is usually a result of a small steam leak. The leak would be undetected on the expansion

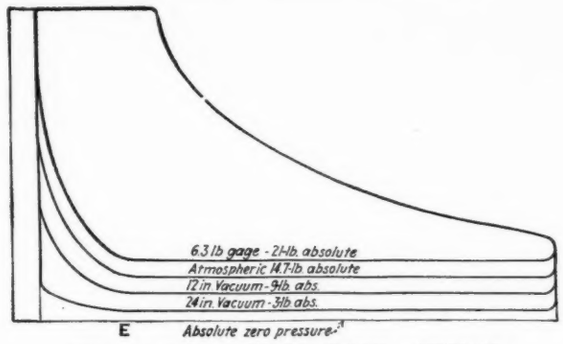


FIG. 9. FINAL COMPRESSION DEPENDS UPON BACK PRESSURE CARRIED

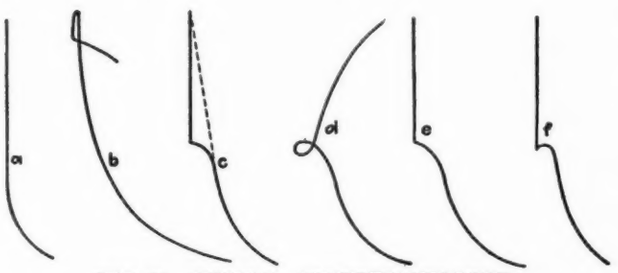


FIG. 10. TYPICAL COMPRESSION LINES

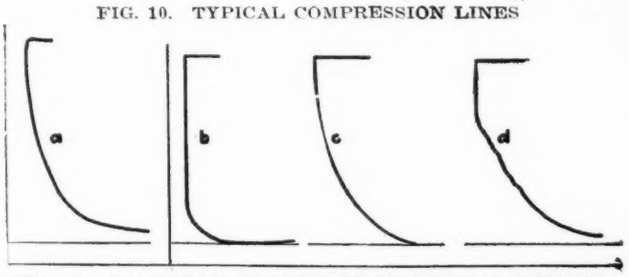


FIG. 11. COMPRESSION LINES FROM TYPICAL ENGINES (a)—High-speed single-valve engine. (b)—Corliss engine. (c)—Four-valve engine. (d)—Throttling slide-valve engine.

line, since the volume in the cylinder would be so great as to show only a slight change in the expansion line. During compression the volume of steam is small and a loss not detectable during expansion might cause a decided decrease in the compression pressure. A serious leak may cause the pressure to remain constant during the final part of the stroke and if the steam valve opens promptly, the curve will end with the sharp admission line at *e*. With late admission the curve may develop a loop as at *d*. The piston has reached dead center, and starting on its next stroke by increasing the cylinder volume to be filled with the clearance steam, causes the pressure to drop sharply until the steam valve allows live steam to enter.

A sharp curve, as at *f*, it is generally agreed, is the result of cylinder condensations which may also cause the shape of the line at *c* and *e*. The compression of the steam raises its pressure and temperature. If the heat lost to the cylinder head is large, the temperature of the steam may drop below the saturation point corresponding to the existing pressure. Part of the steam condenses and the pressure drops sharply. It is possible to determine if the drop is due to condensation or to a leak by running the engine on air. If a leak is the cause, the drop will still show up since air will leak as readily as will steam. However, condensation does not occur with air, and the sharp curve will disappear when using air unless a leak be the cause.

The compression line should closely approximate the $pv =$ constant curve. It is hard to compare, since the length of the compression is by no means large. The engineer can, in most cases, detect any marked differences. In Fig. 11 are shown typical compression lines from engines of various designs.

Esterline-Angus Concentration Meter Is Convenient

The need for a concentration meter in the boiler room comes from the fact that, even in a plant where there are water softeners or where compounds are used directly in the boilers, the continual accumulation of salts left in the boiler by the evaporation of the water causes priming and corrosion unless the boiler is blown down before the accumulation of salts goes too far; and the operator has no way of knowing when that point is reached unless he can measure the concentration of the water in the boiler. Without some way of testing the water, therefore, he naturally keeps on the safe side by blowing down oftener than he would find necessary if he could know the exact conditions.

To give him a means of keeping track of this condition, the Esterline-Angus Co., of Indianapolis, Ind., has put out a meter that makes it easy to test a sample of the boiler water or the feed water in a short time. This meter, which is shown in Fig. 1, is of the portable type, with a little cup on one side into which the water to be tested is placed. There are two electrical contacts in this cup, so that electricity is made to pass through the water. In the same circuit with these con-

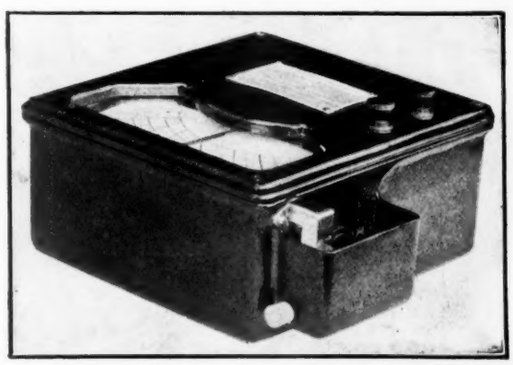


FIG. 1. A QUICK-READING CONCENTRATION METER Water to be tested is put into the small cup at the side, connection is made to an alternating-current circuit, and the concentration of the water is read direct.

tacts is a direct-reading ohm-meter movement that indicates the degree of concentration of the water, since the electrical resistance of water decreases as the concentration increases. The meter scale, of course, is marked, not in ohms, but in degrees of concentration.

The meter is operated by alternating current, but its accuracy is not lessened by changes in voltage. The meter element is of the induction type, mounted on a ball bearing without pivots or jewels.

The chart in Fig. 2 gives a daily record of the concentration of the boiler water and the hardness of the makeup water over a six-weeks test at the Anderson Power House of the Union Traction Company of Indiana. Starting with new water, the concentration increased up to the sixth day, when the boiler was blown down. After that it was blown down three inches of

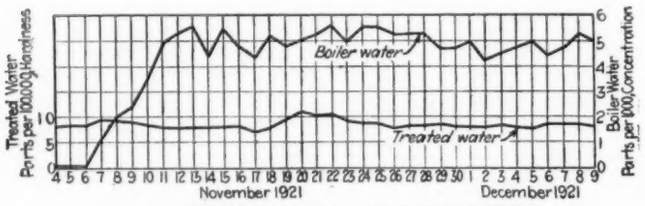


FIG. 2. RECORD OF A SIX-WEEKS TEST
The boiler was blown down three inches of water each day, which was enough to keep the concentration below the point of priming.

water once a day. The chart shows that that amount was enough to keep the concentration below the point of priming. Blowing down more than that would, of course, have been unnecessary and wasteful.

Wattmeter Used as Reactive-Factor Meter

A synchronous motor in an industrial plant was equipped with a three-phase reactive-factor meter which enabled the operator to adjust the power-factor to unity, as was necessary whenever there were wide variations in the load. This adjustment on a synchronous motor is made by varying the field rheostat. One day this meter failed to respond to changes in the field rheostat, but indicated steadily at about 20 per cent reactive factor lagging. Suspecting the failure of a meter coil, the maintenance man proceeded to check his suspicions by various voltage tests.

Referring to schematic diagram of connections, Fig. 1, there are three voltage coils, *A*, *B* and *C*, connected in star, in the meter; and each coil is in series with a 1,000-ohm resistor, *D*, *E* and *F* across the three-phases.

The first test was to connect the voltmeter across points 1 and 2, 2 and 3, and 1 and 3 to see that the

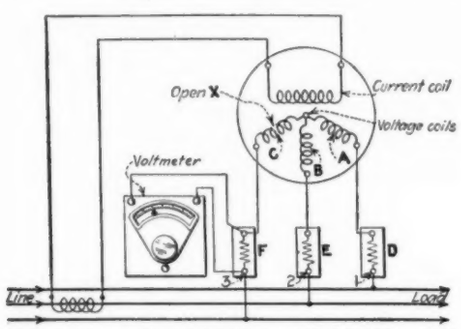


FIG. 1. CONNECTION DIAGRAM FOR THREE-PHASE REACTIVE-FACTOR METER

three-phase voltage was applied. The next test was to remove a wire from *D* and insert the voltmeter in series with the resistor. It gave a deflection showing that current was flowing through the meter and through the resistor. The connection was replaced and then the voltmeter connected in series with resistor *E*. This also gave a deflection, thus indicating a continuous circuit in this resistor and the internal coil to which it was connected. This connection was replaced and the voltmeter inserted in series with resistor *F*, which gave no deflection, thus indicating that there was an open circuit in either the resistor *F* or coil *C*.

After replacing the connection, the voltmeter was connected directly across the resistor *F*, but it did not deflect, thus showing that the current was not passing

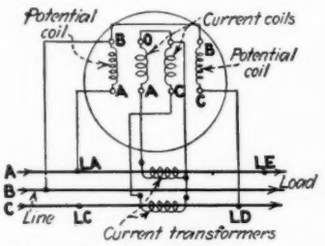


FIG. 2. POLYPHASE WATTMETER CONNECTED TO READ WATTS ON THREE-PHASE CIRCUIT

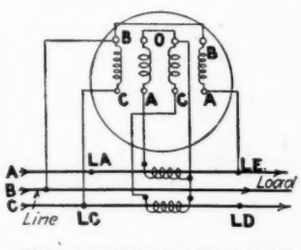


FIG. 3. POLYPHASE WATTMETER CONNECTED TO READ REACTIVE COMPONENT ON THREE-PHASE CIRCUIT

through the meter windings. If it had deflected, it would have shown that the resistor was open-circuited, as the current must pass through the meter in order to actuate the voltmeter. That the meter coil was open-circuited was confirmed by connecting the voltmeter across the meter terminals of coils *B* and *C*, when the deflection of the voltmeter showed that the current was passing through the resistor. The meter was then removed and sent to the manufacturer for rewinding.

It was desirable that some means be provided for indicating the power factor or reactive factor of the synchronous motor. There was available a polyphase wattmeter, and this was connected in to read reactive component. Fig. 2 shows the standard connections for reading watts, while Fig. 3 shows the connections for reading reactive factor. It will be noticed that by simply changing one potential lead from *LA*, Fig. 2, to *LC*, Fig. 3, and the other potential lead from *LD* to *LE*, the wattmeter reads reactive component of the motor's kilovolt-amperes.

By the use of a constant the actual reactive component could be indicated, a lagging component deflecting to the left of zero and a leading component deflecting to the right. This was not necessary, how-

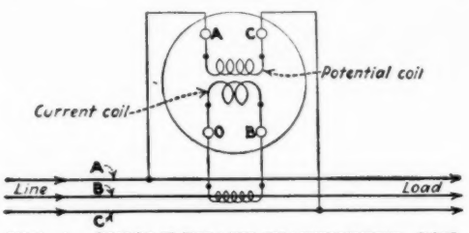
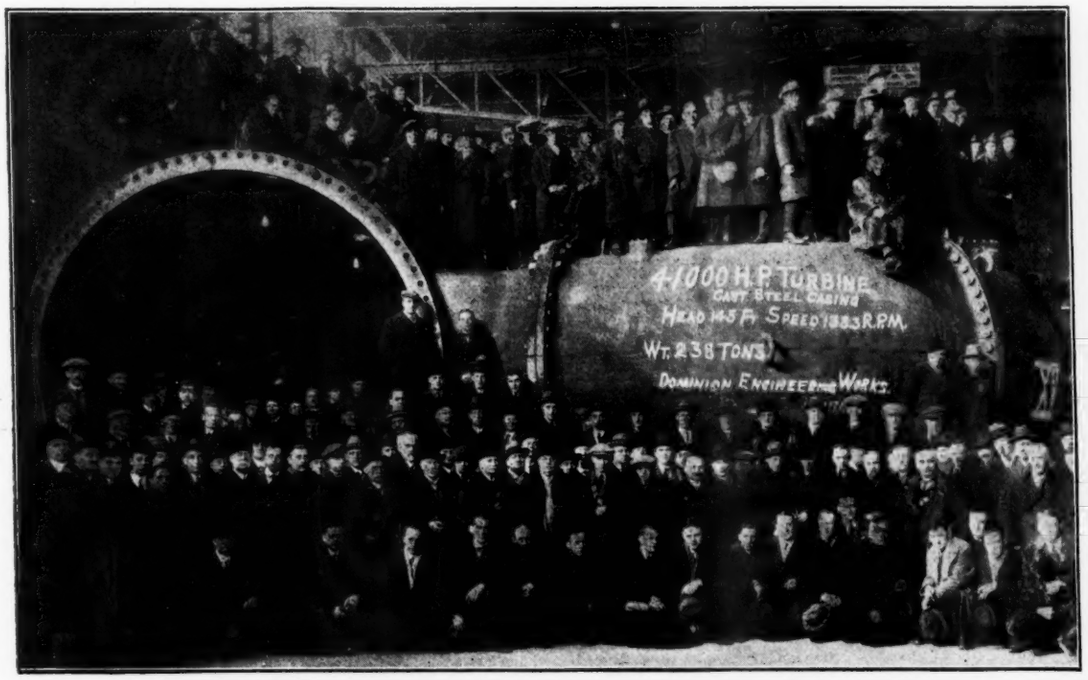


FIG. 4. SINGLE-PHASE WATTMETER CONNECTED TO READ REACTIVE FACTOR ON THREE-PHASE CIRCUIT

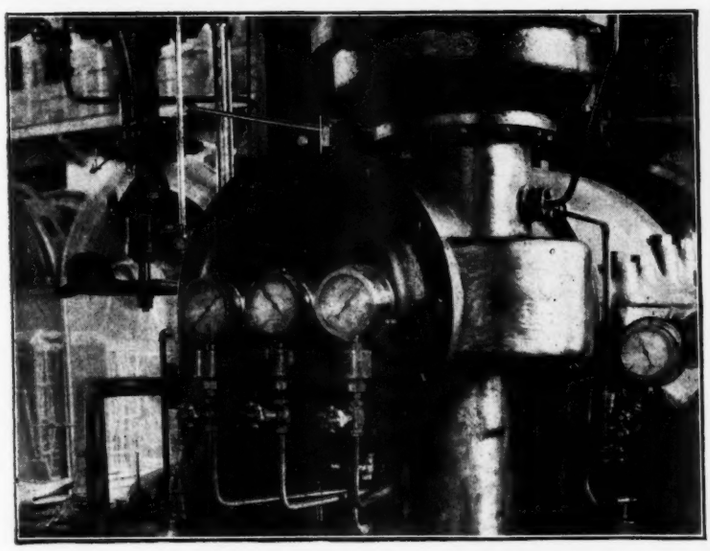
ever. It was enough to know that when the meter indicated zero, the reactive component was zero. When the reactive component was zero, the reactive factor must be zero and the power factor 100 per cent, therefore the field rheostat was simply adjusted to make the meter read zero.

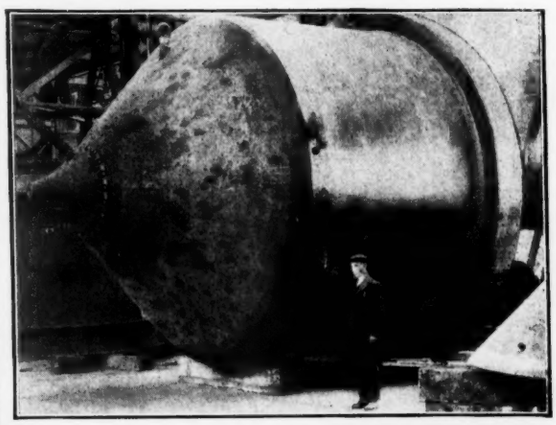
Fig. 4 shows how a single-phase wattmeter can be connected in place of the reactive factor meter and will indicate zero reactive factor when this condition prevails.



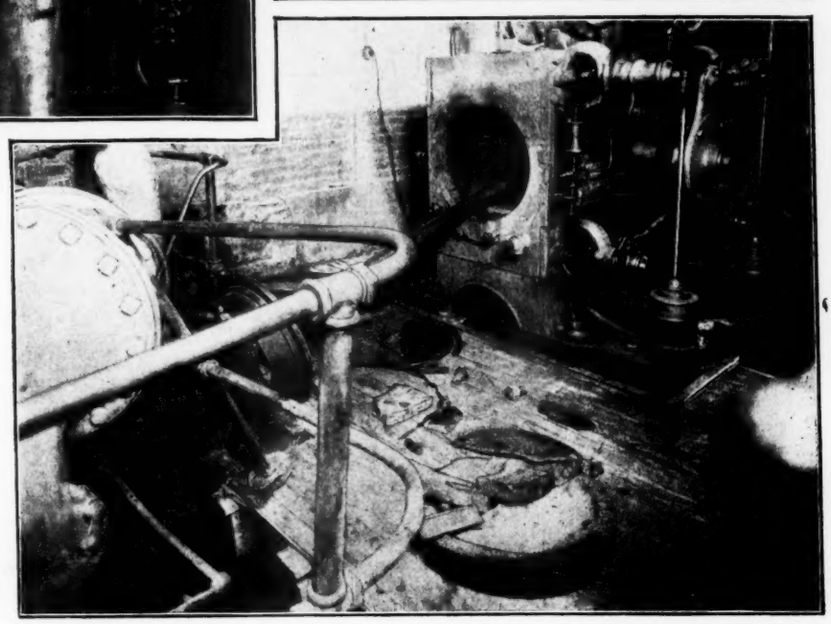
This is the casing for the biggest water turbine in the world. (The Queenston-Chippewa turbines have greater capacities — 55,000 hp. — but are smaller, operating under a higher head at a faster speed.) Grouped about and on it are members of the Engineering Institute of Canada at their recent annual convention. The turbine is for the Shawinigan development, in Canada, was designed by J. P. Morris engineers and is being built by the Dominion Engineering Works, of Montreal. Below is the cast-iron plunger for its 20-ft. Johnson valve.

□ □





Can you see the nickel? It is there, a five-cent piece balanced on the edge of the cover plate of this 20,000-kw. steam turbine while the machine was carrying a 5,000-kw. load. The nickel was in place during synchronizing. The unit was recently put into service at the Hauto Plant of the Pennsylvania Power and Light Co.



Right—The piston rod came loose from the crosshead, and the piston knocked off the cylinder head and did considerable other damage to nearby equipment. The cause was the breaking of the gib which, together with the cotter key, held the piston rod to the crosshead; the cotter dropped out, and steam in the crank end of the cylinder drove the piston out through the head end. It has not been determined why the gib broke.

Courtesy of "The Locomotive," Hartford Steam Boiler Inspection and Insurance Co.

The Cause of Emulsions in Lubricating Oils

By W. F. OSBORNE*

WHEN a highly refined hydrocarbon oil and distilled water are mixed, they will promptly separate cleanly. The oil will usually appear slightly cloudy for a time, and the water will also have a cloudy appearance, which will gradually disappear.

The rate at which the water and the oil separate will depend on the density and viscosity of the oil. Oils that have the lowest specific gravity will rise to the top of the water the most quickly, on account of the greater difference in the density of the oil and the water. A 28 Bé gravity oil will separate somewhat more quickly from water than a 27 Bé. gravity oil.

As the temperature is raised, the density of hydrocarbon oils drops more rapidly than does the density of water. In heating an oil from 60 to 150 deg. F., the decrease in density amounts to about 7 per cent, whereas heating water from 60 to 150 deg. F. decreases its density only 1.1 per cent. For this reason it is possible to get a somewhat more rapid separation of oil and water at higher temperatures than at low.

Separation of Water and Oil

A factor having a much greater influence on separation from water, is the viscosity. At equal temperatures and with equal gravities the oil having the lower viscosity will separate the more quickly from water. Water globules mixed with oil globules can find their way to the bottom of the vessel more quickly when the oil globules are easily moved aside, as would be the case for a thin-bodied oil. This is especially pronounced when the amount of oil is large as compared with the amount of water or when the oil contains only a small percentage of water.

This property is taken advantage of when it is necessary to separate water and oil mixtures by heating the mixture, thereby reducing the viscosity of both the oil and the water. Raising the temperature of water from 60 to 150 deg. F. reduces its viscosity by about 55 per cent. Raising lubricating oils through the same temperature range decreases their viscosities from 90 to 97 per cent depending on the original viscosity and the character of the oil. It would appear that oils having steep slopes to their viscosity temperature curves would be particularly suited to the lubrication of machinery where large quantities of water are likely to become mixed with the oil, on account of the fact that a relatively small temperature rise will produce a considerable decrease in viscosity.

The property of separation possessed by mixtures of hydrocarbon oils and water holds good if the oils are highly refined and if the water carries no impurities. The addition of any foreign material to the oil or water greatly decreases the ability to separate. When the oil and water fail to separate, an emulsion is formed. This emulsion consists of globules of oil and water thoroughly mixed together, and may be regarded as a temporary or permanent emulsion, depending on whether it will respond to treatment intended to break it up.

For certain purposes emulsions are very desirable, and considerable effort is expended to make them permanent. Soluble cutting oils which are emulsions of water and compounded petroleum oils are very valuable for the lubrication and cooling of cutting tools. Such emulsions contain a relatively large percentage of water and a small amount of oil.

The emulsion formed in the circulating system of a steam turbine is very undesirable. It plugs up oil lines, prevents the proper settling out of impurities and by carrying the impurities to the bearings, causes the latter to become heated and to wear. Whenever possible, the occurence of emulsions in lubricating systems should be prevented.

Impurities in Oil

Many impurities, when added to either the oil or the water, will cause the formation of a certain amount of emulsion. Sulphates, chlorides, finely divided clay and earthy matter, organic matter, sewage, etc., contained in the water usually available in power houses increase the tendency of the water to remain mixed with the oil and to prevent proper separation. Alkaline boiler compounds which get into the oiling system through condensed steam are a frequent cause of emulsions, owing to the action of the alkali on certain constituents of the oil, particularly after it has been used for a time. Animal and vegetable oils, certain acids and alcohols tend to increase the formation of emulsions.

Effect of Compounding

Lubricating oils are made in several grades depending on the service to which they are to be put. Some oils are so refined that they will separate from water more readily than others. The tendency of certain other oils to hold water in suspension is due to the presence of certain compounds which were not entirely removed during the refining process. Unless the oils are to be used under conditions where they will become mixed with water, these compounds do no harm and have no deleterious effect on the actual lubricating value of the oil. They can be removed by suitable treatment during refining, and oils so treated will separate satisfactorily from water except under extremely adverse circumstances.

Various attempts have been made to devise a satisfactory test to determine the quality of an oil with respect to its freedom from emulsion-forming tendencies, but so far little success has been met with. The Bureau of Standards has prepared a test that shows the rate at which equal quantities of oil and water will separate when mixed together under certain conditions. The name demulsibility has been given to the property of the oil which this test reveals. Unfortunately, the rate at which two oils might separate from water does not necessarily indicate the completeness of that separation. For instance, an oil that separates very rapidly from the water might leave at the end of the test a thick film of emulsion between the water and the oil. Certainly, such an oil would not be as suitable as another

*Supervisor Manufacturers' Service, Texas Co.

oil which might separate a little more slowly, but which would leave no film of emulsified oil at the end of the test.

The American Society for Testing Materials has not yet developed a satisfactory test for the emulsifying qualities of an oil, so that the engineer will still have to depend largely upon practical trials to determine whether an oil will give satisfactory service.

A simple test used by many engineers is to fill a four-ounce bottle with equal parts of water and oil. The bottle is shaken vigorously for one minute and set aside to settle. At the end of one hour the water and oil should be completely separated and there should be no film of yellow emulsion between them. When comparing two oils, be very careful to have the temperatures of the water and oil in each case exactly the same.

Q. I am using the best grade of turbine oil I can secure and still have trouble with emulsions occasionally. What can I do to prevent this?

A. Emulsions can frequently be prevented by raising the temperature of the oil in the circulating system. If the oil leaves the cooler at too low a temperature, it may not get hot enough on going through the bearings to lower its viscosity to the point where the water can separate. A temperature of 160 to 175 deg. F. will improve the separation of water considerably, and if the oil originally has the proper viscosity, no trouble will be experienced with bearings running at this temperature.

Emulsions can be broken up sometimes after they are formed, by heating the oil to a fairly high temperature and allowing it to come to rest in a suitable tank so the water can settle out. Of course the best plan is to eliminate the source of the water and the impurities that it contains, but this cannot always be done. Sometimes emulsions are caused by a very rapid rate of circulation and can be eliminated by increasing the capacity of the system, in this way lowering the velocity of the oil and the amount of agitation, thereby giving it more time to separate from the water.

Handling Ashes by Sluicing

Calumet and Hecla Power Plant Reduces Ash-Handling Costs by a Hydraulic System of Ash Disposal

BY C. C. LANCE*

ALTHOUGH the power plant of the Calumet & Hecla Mining Co. at Lake Linden, Mich., has been completed for several years and is not of the most recent design, it has a number of interesting features. Power is generated for the many copper mines surrounding the plant and for running a large pumping plant which supplies water to the mills and mines and also to several small mining towns. Current for lighting is also furnished.

In the boiler room there are twenty-two 5,500-sq.ft. water-tube boilers set in pairs and divided up into four batteries, three of six boilers and the other of four boilers, with space for an additional twin setting of the same size.

The coal-handling facilities in the boiler room of this plant consist of an endless bucket conveyor arranged to elevate coal from the coal crusher, where it is crushed into stoker size, into overhead bunkers feeding into the stokers on the individual boilers. A sufficient supply of coal to run all plants for practically an entire season is laid in. Coal is brought to the boiler room in standard-gage hopper cars and dumped into the hopper of the coal crusher.

Two men hoist the coal used—one man overhead to arrange the bucket trips to fill the desired bins, the other man to dump the car doors and run the conveyor and crusher. Considerable trouble is experienced due to foreign matter feeding into the crusher with the coal, and it is not at all uncommon to break one or both crusher rolls.

Perhaps the most interesting feature of the plant is the method of handling ashes. Each boiler has a large ashpit underneath with two bottom doors opening into an underground ash tunnel. The ashes are collected in these pits and dumped four times each day. A modification of the sluicing processes used throughout the copper refining is employed for handling the ashes. There are four troughs, one under each battery of boilers. All are built of steel plates with structural supports.

These troughs slope toward their junction points with a pitch of ¼ in. per foot. The two troughs on the same side of the boiler room join into a single trough which continues to a central tank about 5 ft. wide, 7 ft. long and 5 ft. deep. An 8-in. suction line connects to this tank two 8-in. motor-driven belt-connected centrifugal pumps which remove the water and ashes. One pump maintains the service, and the other is kept for emergency use, as the ashes wear out impellers about every seven weeks and liners every twenty-one weeks. Thus one pump is available while the other is being repaired.

When operating this system, two men are required, one to open the ash doors under the boilers and the other to start up the centrifugal pump and break up the clinkers so that they will pass through screen bars into the suction tank. Before starting to dump ashes, water is turned into the sluicing trough at the high end by opening an 8-in., quick-opening gate valve, which delivers a suitable volume of water at about 45-lb. pressure. When a flow of water has been established, the ashpit doors are opened singly and the ashes are washed into the suction tank and pumped out by the centrifugal pump through an 8-in. pipe line.

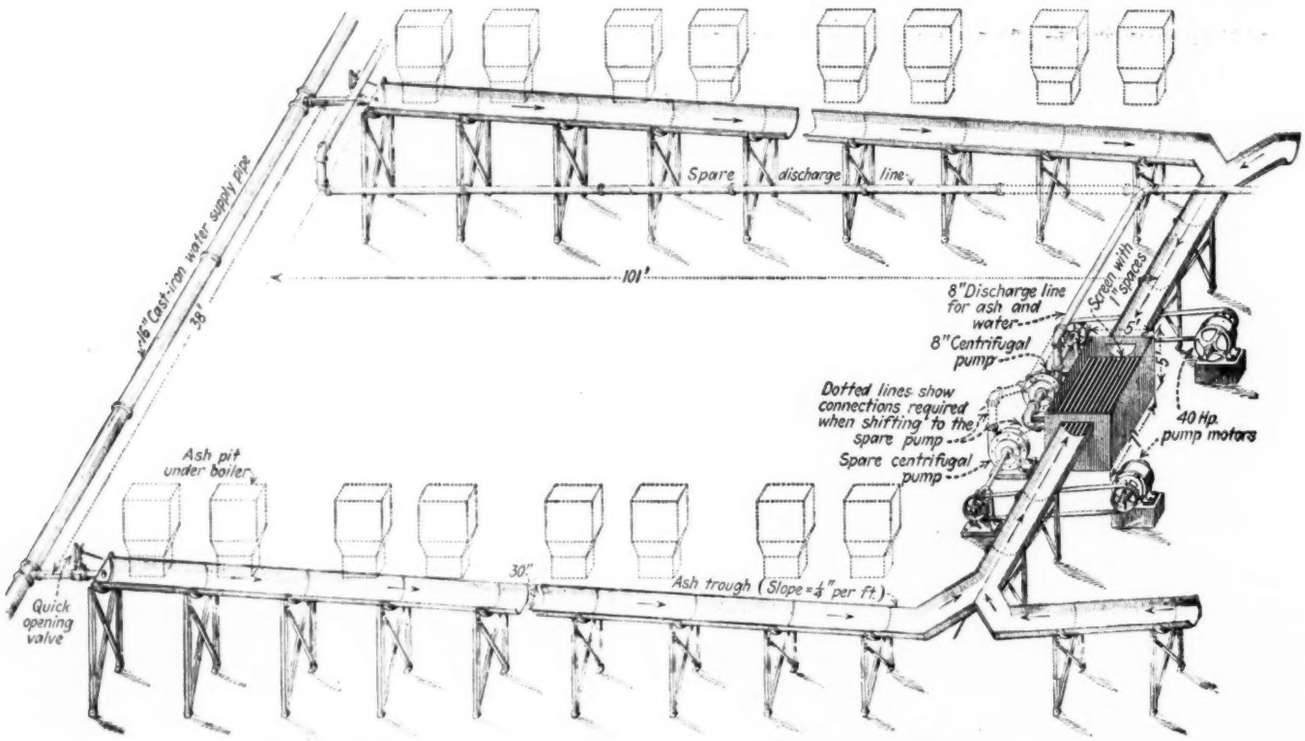
With the entire plant running full, only four men are needed to dump all the ashes and dispose of them, as well as handle coal. At present two men, working eight hours, handle the coal and ashes from the six boilers in use by working broken shifts. Four dumps are made per day, taking about two hours each.

When the stamp mill was running, the ashes were piped to a 60-ft. sand wheel used for elevating mill tailings and were disposed of with other mill sand and refuse, but at present the ashes are run through about

*Engineer, National Boiler Washing Co.

600 ft. of 8-in. pipe and used for filling some low ground. The sluicing troughs were installed in the ash tunnels and elevated as much as possible at the far ends to prevent the ashes and clinkers from sticking in the troughs, but the slope is not so steep as it really should be to move the ashes into the suction tank without assistance from the men. It is found necessary to start each lot of ashes dumped into the trough by pushing with a shovel, and the same help is required at the bends where the troughs join.

Local conditions, of course, are responsible for this, and the installation was made in the best way possible, but in planning a similar installation care should be taken to eliminate bends and turns in the troughs and to increase the grade to approximately ½ in. per foot.



ISOMETRIC SKETCH OF THE ASH-REMOVAL SYSTEM IN USE AT LAKE LINDEN

If this is done, no hand labor will be required other than for dumping the ashes into the trough.

Water requirements vary with the size and quantity of the ashes and clinkers handled, but with all boilers in operation and making approximately 45 or 50 tons of ashes a day, it should not require over 250,000 gal. of water to dispose of the ashes.

The pumps used for removing the ashes from the tank and disposing of them, are of the type known as sand pumps. The specifications under which they were designed called for two 8-in. centrifugal pumps with capacity to handle 100 tons of ashes and 5,000 tons of water in 8 hours at 750 r.p.m. against a head of 78 ft. Each pump is driven by a 40-hp. motor, belt connected. The speed is cut down to 546 r.p.m. instead of that specified. This reduction in speed lowers the efficiency but prevents breaking the suction, which would be the case if the pumps were used to their maximum efficiency, as the quantity of water required is not constant and varies with the number of boilers in operation.

Replacement of impellers and pump liners is not a matter of moment with the company because it has a foundry and casts these parts of chilled gray iron. This is also true of the various elbows and fittings used in the pipe line carrying away the ashes.

With the peculiar conditions existing at Lake Linden, in that there is no objection to refuse and mill tailings being dumped into the lake, the method in use seems to be the best that could be devised. There is a question, however, as to whether this would be true with conditions as they exist in most large central power stations.

For one thing there are few plants except those located close to their own water supplies that could afford to use the water needed. Even a waterside station would probably find the cost of pumping water for this purpose a matter to be carefully considered, and there are few plants that could dispose of ashes with so little difficulty.

For a large plant the most efficient arrangement of an ash-disposal system of this type would include an overhead steel or concrete tank having a separating screen, into which pumps would elevate the water and ashes. The water could be drained off and the ashes dumped into railroad cars after a sufficiently large quantity had been pumped into the tank.

The surface wear of the steel troughs or ash launders is practically nothing at all, those at Lake Linden having been in use for over three years without showing any wear other than a slight smoothing of the inside surface. As far as could be ascertained, the maintenance cost of this ash-handling system has been chiefly the cost of the liners, impellers and cast-iron pipe fittings and the labor required to replace them.

When the mills and mines are in full blast, the water consumption is large, and in addition, a large amount of water is pumped to several towns from the central pumping plant operated by the company. This reduces the cost per thousand gallons to a low figure.

Ice-Making With the Unaflow Engine

Remarkable Economy Within Easy Reach—Electric-Motor and Steam-Engine Operation Compared

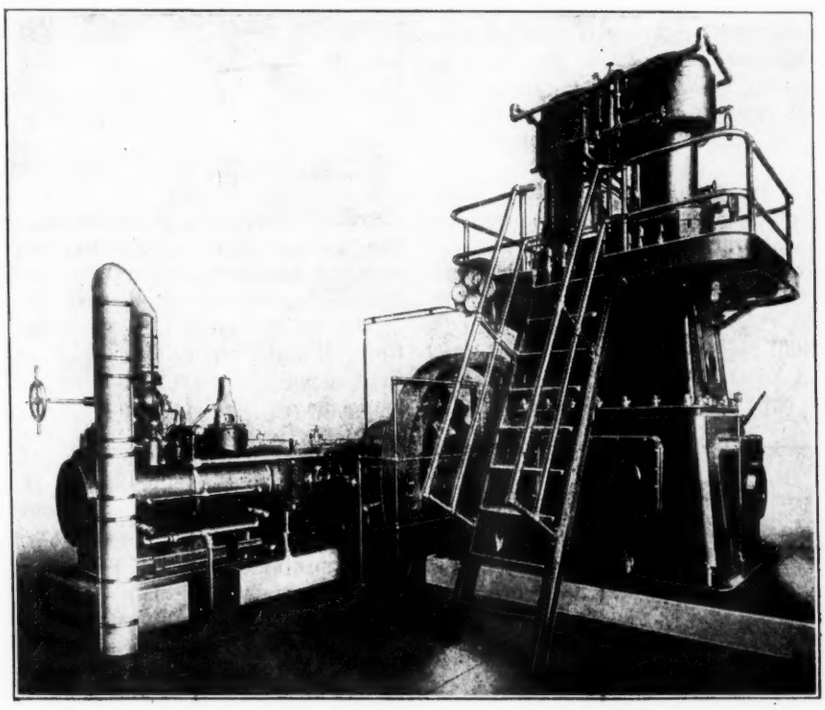
By STERLING H. BUNNELL*

NOT so long ago the best-designed and operated can ice-making plants in the United States were producing little over six pounds of ice per pound of coal, with an occasional seven as the best performance possible. In Europe, where ice is a luxury and coal expensive, the production of ice per pound of coal runs considerably higher. But the European never wastes good coal to distill and reboil the water to fill the ice cans. White ice is good enough for his purposes, and impurity makes no difference when ice is never put into anything to drink. America, on the contrary demands ice made from pure healthful water, and competition requires artificial ice to be clear, therefore it must be free from sediment.

The ice plant using condensed steam for filling the cans has to produce about 1½ tons of steam for each ton of ice made. Allowing 15 per cent of the steam for the auxiliaries, the rest is enough to run the main compressor engine at 33 lb. of steam per hour per horsepower. There is no possible object in using less steam per horsepower, as in any case the 1½ tons of steam must come from the boiler. With the Corliss engines generally used in ice-making plants, it was common to find a large live-steam makeup blowing direct to the condenser, while the engine might be doing its horsepower on 28 or better.

The raw-water system changed the whole steam-engineering practice in connection with ice-making. With good water available in every city there is no object in making distilled water ice, and the whole troublesome system of condensers, reboilers, skimmers and filters can be done away with. Immediately, the production of steam can be reduced to that necessary for producing power; or the whole plant may be operated by electricity taken from the central station. A motor-driven raw-water ice plant can be operated with very little attention from a skilled man, and as this fact becomes better known, the number of small plants for local supply is rapidly increasing.

The power required by an ice plant varies, of course, with the temperature of the condensing water and with other conditions that depend on the season and weather; but some comparisons can be made by taking the same reasonable or average conditions throughout. In a motor-driven plant, in moderately hot weather, 40 to 45 kw.-hr. will operate the main compressor to produce a ton of ice from city water, and the auxiliaries will require 20 to 25 per cent more. The required amount of power is therefore 50 to 55 kw.-hr. per ton of ice. If the price is two cents per kilowatt-hour, the power cost is 1 to $1.10 per ton of ice made. The total of labor and other costs will be very much less than the cost of power, even though the plant may be as small as 50 tons in capacity.



UNAFLOW ENGINE DRIVING AMMONIA COMPRESSOR

Modern compressor design, including light-weight valves, allows direct connection to medium-speed unaflow engine.

The large steam-operated ice plant has far greater possibilities of economy. With the raw-water system the cooling starts generally at not above 70 deg., so that a ton of ice can be reduced for 1½ tons of refrigerating capacity, requiring only about 63 i.hp.-hr. by the main steam engine. This power should be supplied by a condensing unaflow engine at a water rate of approximately 13 lb. per i.hp., or a total of 819 lb. of steam per ton of ice. If the evaporation is 9 to 1, a ton of ice can be produced by 91 lb. of coal, at the rate of 22 lb. of ice per pound of coal. But the auxiliaries will cut down this performance considerably. If they require 15 per cent of the power of the main compressor and are motor-driven by current from a generator operated by a unaflow engine, their equivalent steam consumption will be about 155 lb. The ice-production rate is thereby reduced to 18½ lb. of ice per pound of coal. Taking $5 per ton as the price of coal, the coal cost per ton of ice is only 27 cents. For the old

*Consulting engineer, New York City.

can-ice making plant at 6 lb. of ice per pound of coal, the fuel cost would be almost three times as great.

With all due respect to the specialists in refrigeration the planning of the details of an ice plant to produce 18 lb. of ice per pound of coal is distinctly work for engineers with the widest experience in modern power-plant practice. It is easy enough to produce a horsepower on little more than a pound of coal, if the main engine or turbine alone is taken into account. The auxiliaries, however, play havoc with paper economies based on large-unit performance. The large power stations with remarkable performances in coal per kilowatt-hour are the ones in which auxiliary operation and the saving of every possible heat unit have been most carefully worked out by the designers and followed up by the operators. The same broad knowledge and experience can provide an ice-making equipment that will produce in ordinary hot summer weather, ice at the rate of 18 lb. per pound of coal. Under favorable local conditions, such as a plentiful supply of cool water at easy pumping level, the 18-lb. figure may be considerably surpassed.

Federal Pressure Switches Use Mercury

A line of pressure-operated electric switches recently designed, uses the familiar principle of the Bourdon tube, as found in steam and other pressure gages, together with a novel type of switch that opens and closes the circuit by the movement of mercury in a sealed glass tube.

This tube is shown in two positions in Fig. 2. When it is tipped as shown at the top, there is no connection between the two electrical contacts, and when it is tipped the other way, the mercury slides along to the right-hand end, makes a connection between the two points and closes the circuit. The tube is filled with a gas that is inert, so that there will be no chemical reaction, and that is non-conductive, so that arcing will be reduced. Any mercury that is vaporized by the arcing will be condensed on the walls of the tube and so will not be lost. The makers call particular attention to these facts as indicating that the device should have a long life.

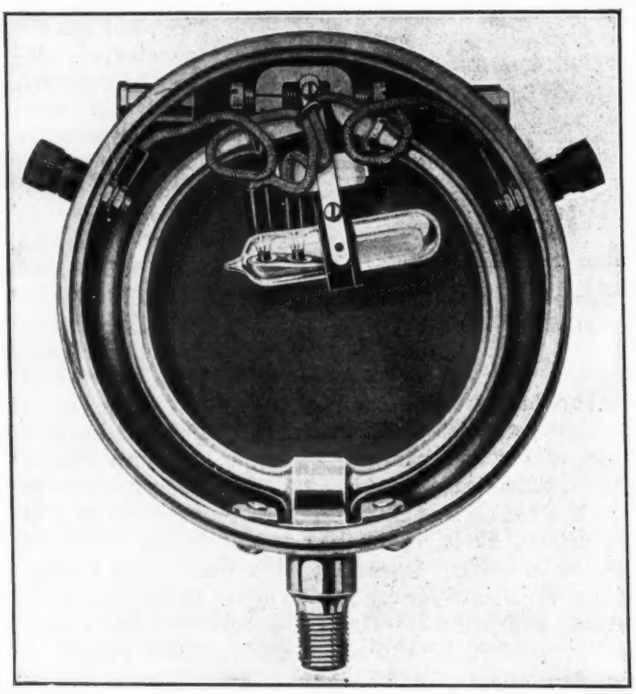


FIG. 1. INSIDE OF THE COMPLETE INSTRUMENT

Increasing pressure forces apart the ends of the Bourdon tube, tilts the glass mercury tube and so breaks the circuit.

Fig. 1 shows how the glass tube is tilted by changes in pressure. The Bourdon tube (the hollow, circular tube with opening at the bottom of the instrument) is much the same as in the usual steam gage, but in this case it is connected by levers to the mercury tube instead of to a pointer. It may readily be seen that an increase in pressure in the Bourdon tube tends to spread the two ends of it, tipping the glass tube so as to let the mercury run down into the right-hand end and open the circuit. A decrease of pressure does just the opposite.

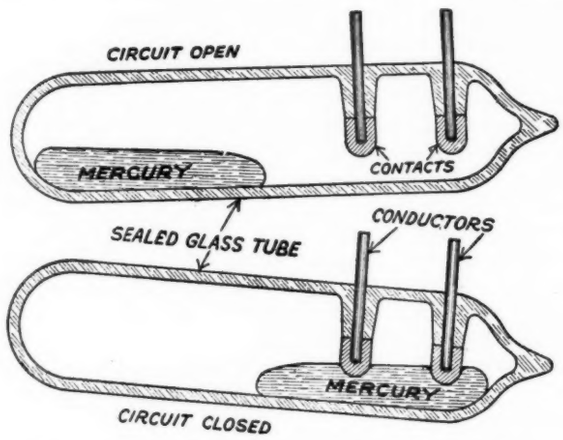


FIG. 2. THE TUBE THAT MAKES AND BREAKS THE CIRCUIT

Slight tilting of the tube rolls the mercury from one end to the other, thus connecting or disconnecting the two electrical contact points.

The Federal Gage Co., of Chicago, and the Marshalltown Manufacturing Co., of Marshalltown, Iowa, are the makers of the switch, to which they have given the trade name of "Mercoid." They have standardized about 35 Bourdon tubes with pressures from 7 oz. to 30,000 lb. to meet different conditions of service. The finest covers a 7-oz. maximum pressure and is able to operate on a minimum range of 0.2 oz. At the other extreme is a tube for pressures from 0 to 30,000 lb., with a minimum range of 100 lb.

The mercury tube has been standardized in three forms, of which the simplest is shown here. In a second form there is a pair of contacts at each end of the tube, and the third form is similar except that the two middle contacts are joined. These three kinds of tube can be used in multiple to give twelve additional combinations of circuits. The maximum rated working capacity of all these switches is 10 amperes at 110 volts or 5 amperes at 220 volts.

Movements for the switches have also been standardized in three forms, with 30 possible combinations. The one described here is the sensitive type. A second form is the stop-latch type, which needs larger ranges of pressure but will stand harder use. A third type uses the relay principle and so allows for sensitive adjustment in spite of severe service conditions, such as are caused by vibration.

These three types of movement are supplied with the pressure and range adjustable, with a set range and the pressure adjustable, or with no pressure or range adjustment whatever. They are made also as safety devices, working in only one direction and then locking; they are then reset by hand.

EDITORIALS

Meeting the Coal Situation

ON JANUARY FIRST there were in stock nearly fifty million tons of bituminous coal in the hands of consumers. Almost every week since small but appreciable additions to these stocks have been made. The maximum possible interruption of coal supply that can occur due to the threatened strike is estimated on the basis of official figures to be a reduction of bituminous output to about six million tons per week. During the summer months bituminous consumption under present conditions will certainly not exceed eight million tons per week, and probably be under that figure materially. It takes no complicated calculations to show, therefore, that for the country as a whole, the situation does not immediately spell hardship for industry.

He who has been dilatory in purchasing coal can take some comfort from these facts, but he may well take warning too. The coal stocks, though in general adequate, are not uniformly distributed. There will be local cases of shortage or cases of individual plants having difficulty in securing regular fuel supply. However, these cases will not justify payment of any exorbitant premium for coal. It is to be hoped that purchasers who find themselves in that unfortunate position can meet the situation without panic. Certainly, they should not allow temporary or local shortage to result in runaway market conditions.

Again let it be emphasized in the mind of every purchaser that though too late to stock coal now, it is by no means too late to keep a sane attitude toward the pending situation and decline even at the expense of some hardship to pay extortionate prices for fuel.

Power-Factor Correction Requires Careful Study

LOW power factor is the besetting evil of most alternating-current systems. Where a large percentage of the load served is induction motors, unless means are taken to correct the power factor it cannot, even under the best of conditions, be much above eighty per cent. If there are a number of large, slow-speed, lightly loaded motors on the system, the power factor may be less than fifty per cent. The latter, fortunately, is the exception and not the rule. However, power factors as low as sixty per cent are not uncommon, which means that forty per cent more generator, transformer and line capacity is required to supply a given kilowatt load than at unity power factor. To this may be added many other bad effects of low lagging power factor, such as decreased efficiency of power equipment, poor voltage regulation and so forth.

For many years power-plant engineers have recognized the advantages to be gained by maintaining the power factor of alternating-current systems at a high value. However, it remained for the vastly increased cost in power, experienced in recent times, to emphasize the necessity of action to remedy the low power factor of industrial power loads, as a means of eliminating at least part of the increase in power-production cost. Although considerable has been done toward this end, it can hardly be considered much more than a beginning. This may be looked upon as a danger point, since if proper methods are not employed in correcting the conditions, large and unnecessary expense may result without producing the desired effects. The installation of a synchronous motor or a static condenser on a low-power-factor industrial load may correct the condition at the power plant or transformers, but the reason for the low power factor still exists. Under such a condition the corrective device must be continually in service if high power factor is to be maintained.

Among the various means of correcting low power factor is the rearrangement of the induction motors in the plant to get them carrying somewhere near their normal load. This generally necessitates the purchase of one or two small motors and the disposal of an equal number of large machines. However, in systems where rearrangement can be taken advantage of, the cost of the change is usually low. If the power factor can be improved in this way to near the motor's normal full-load value, a correcting device may not be required; or if it is, then one of a size considerably under that required originally will serve the purpose. In many installations it is impossible to keep the motors loaded on account of fluctuating load conditions. Unless these loads can be driven by synchronous motors, some means must be provided for correcting the power factor if it is to be maintained at a high value. No two installations are alike, and what may be the ideal solution in one case may be unsatisfactory in another, therefore each should be carefully studied before recommendations are made.

The solution of the power-factor problem cannot be satisfactorily arrived at by any one means, but by an intelligent application of the various schemes available. Where a rearrangement of motors may give the desired results in one case, the installation of a synchronous motor, static condenser, synchronous condenser, or a combination of some of these will give the proper solution in another.

Boiler-Furnace Development

MODERN boiler furnaces are designed to ensure as near as possible perfect combustion of the gases with the smallest practicable amount of excess air. Naturally, these conditions are associated with maximum practical flame temperature provided the furnace volume is sufficiently large to permit complete combustion of the gases before the tube surfaces are reached.

It has been observed that high boiler efficiency is associated with rapid absorption of radiant heat. All recent designs of large boilers embody the idea of absorbing the greatest possible amount of radiant heat by exposing several of the lower rows of tubes to the

full glow of the furnace. The effect of this absorption of radiant heat is to reduce furnace temperatures, and materially prevent the brickwork from melting down.

Many engineers believe that the limit of stoker and furnace development is the ability of brickwork to withstand the high temperatures that may be developed, although many makers of firebrick place the blame on operating abuses. However, hollow blocks are now employed in the stoker side walls to keep these cool and to prevent adhesion of clinker. It has even been suggested that some air should be allowed to pass up along the brick side walls to help cool these down. Cast-iron plates cooled by forced draft have also been employed as side walls to stokers, but these develop difficulties when the boiler is in stand-by condition. Hollow boiler walls have been tried with air circulation through the hollow space to cool the thin inner firebrick lining.

In the Lakeside plant at Milwaukee a screen of water tubes forming a section of the heating surface of the boiler has been added above the ashpit floor so that the absorption of radiant heat from the ash and furnace by these tubes prevents the slagging of the ash. One type of marine boiler has furnace side walls consisting of six-inch square steel boxes suitably connected to the boiler drum and forming a portion of the water-heating surface, thus insuring cool side walls and preventing the adherence of clinker.

With these various conditions in mind, it is logical to ask why side-wall construction with water tubes cannot be employed in our large boiler furnaces and thereby overcome the limitations of brickwork and of further stoker development. The structural difficulties are not serious. Such water-tube side walls can function with the minimum of excess air in the furnace and there will be little adherence of clinkers to such surfaces. Furthermore, the water-cooled metal walls will give no trouble in stand-by condition. The gravest difficulty appears to be the amount of radiant heat that this tube surface would absorb. In fact, so much might be absorbed that the furnace temperature would be lowered sufficiently to decrease boiler efficiency. Low furnace temperatures might be prevented by not employing full metal side walls, but rather walls consisting of alternate tubes and special brick which are cooled sufficiently by the exposed tube sections to prevent any softening. Too low furnace temperatures could be avoided by lessening the radiant heat absorbing surface in the lower tubes above the fire. Such a furnace construction really would amount to the transference of some of the radiant heat absorbing surface from the lower tubes of the boiler to the side walls of the furnace. If the radiant heat absorbing surface is properly proportioned, higher boiler efficiencies should be obtainable, together with better stoker performance.

What It Costs To Store

IN THE power plant there is a great diversity of size, shape or other characteristics of spare parts that must be maintained for emergencies. Everything, from the smallest stove bolt to the largest essential part that is thus stocked, costs the management money not only for its purchase, but also for its preservation. The cost of preservation is not alone that involved by the interest on the investment in the parts, for often it costs much more for the space in which to keep them than it does for the parts themselves.

To keep in stock a few extra sizes of rope is not of any consequence, it might seem, but representatives of the cordage industry in conference with the Bureau of Standards insist that the expense of storage alone of the unnecessary diameters of rope which are now made amounts to between $200,000 and $300,000 per month. This situation is true not alone in the cordage business, but in every branch of industry that makes more sizes or styles of material than industry really needs.

Much good has been done in the past by industrial groups working on the problem of uniformity in style or size of machine parts. Still further progress promises to follow the splendid work that the Bureau of Standards and the Department of Commerce are now urging in the general effort for simplification and standardization of industrial activities. The work cited on cordage is only one of a score of examples of such activity. The work has been extended into many fields already, including commodities of interest to many diverse branches of business.

Power-plant engineers have their part in this movement. Of course, at the outset it will not be possible entirely to eliminate old-style or odd-size parts, for first-class equipment still in use that promises additional periods of valuable service cannot well be replaced all at once by new types simply because the latter happens to be the proposed standard of the future. Nevertheless, progress in this direction can be made gradually.

Whenever additional units are to be installed the problem is met afresh. It is a question whether the newest, most efficient unit of a novel form for that particular plant will be better, or whether another unit of the type already installed would be best. In favor of duplication of existing installation are the advantages of familiarity of the operating force, availability of spare parts and simplicity of general plant operation. On the other hand, in favor of improved designs there is always the possible advantage of greater efficiency, greater safety or greater ease of operation to be considered.

It is a good thing to study each such problem carefully, for one must not be averse to progress simply because there is an unfamiliar element or a novel detail in operation to be met. Conversely, sheer novelty of form, unless supported by real improvement in performance, may mean that many additional spare parts are needed in order that reasonable continuity of operation can be assured.

The way that some power-plant piping systems are laid out and installed it is apparent that the designer felt sure he was never going to have to operate the plant. Probably no other part in the design of plants has received so little attention as the piping, and this has resulted in installations that are complicated, much of the piping arranged so that repairs or changes are difficult to make, and the whole system required much more material than would have been necessary had it been laid out properly.

What next? Recent reports come from Munich that an inventor by the name of Preuckner has developed a process for producing artificial coal from stone, which equals anthracite in heating value. The quality of the coal has not been mentioned. However, if it is of no better grade than some of the war-time coal, undoubtedly this inventor can do all he claims.

PRACTICAL IDEAS from PRACTICAL MEN

Operating Troubles with Duplex Pumps

We are using three pumps alternately, two of which are duplex and one simplex double acting. These pumps are used to take yeast from a box 36x40x12 in. into which it flows from the separators. Neither of the duplex pumps will handle the yeast properly—they operate with short, jerky strokes, but when testing the pumps with water, they operate perfectly.

There is an open vent on one pump and an air chamber and vent in the discharge of the other pump. The simplex pump, which is also fitted with an air chamber, handles the yeast without trouble although it is farthest from the receiver. The suction line to all pumps is 2½ in. and the discharge 2 in. The yeast accumulates a considerable amount of air when being separated, and I have concluded that the box is not large enough to give the yeast a chance to release the air from it.

I would appreciate practical suggestions from readers of *Power* as to what might be the cause of the trouble and what could be done to correct it, remembering that the pumps work perfectly with water and that it is only when pumping yeast that trouble is experienced.

Milwaukee, Wis. ALBERT F. BROWN.

Freezing of Air Pipes in a Raw-Water Ice Plant

Compressed air, at a pressure of about 5 lb. per sq.in., is used in a raw-water ice plant for agitating the water in the freezing cans. In cold weather the engineer has had much trouble on account of the outlet orifices of the air pipes, which are inserted in the cans, becoming frozen up. When this happens, agitation of the water ceases and cloudy blocks of ice result.

The engineer arrived at the conclusion that the trouble was due primarily to moisture in the air. The air was delivered by a rotary compressor, to a double-pipe cooler, where the heat of compression was absorbed and carried away by a current of cold water. The water vapor in the air was condensed during the passage of the air through the cooler. Just beyond the cooler the air passed through a separator with a trap connected to it. This, however, could not be depended upon to eliminate all the entrained water.

In warm weather the temperature of the air was not reduced sufficiently in the cooler to cause trouble from freezing of the moisture, but in cold weather the temperature reduction in the cooler was considerable, and when this reduction was further augmented by the refrigerating effect due to re-expansion of the air issuing from the agitating pipes, the result was a temperature sufficiently low to produce rapid freezing.

To correct the difficulty, it was obviously necessary to keep the air above the freezing temperature. Use of the double-pipe cooler was, therefore, abandoned during spells of cold weather. But this did not mend the trouble entirely. The engineer still had difficulty with the agitators when the temperature of the outside air dropped to a low point.

He finally decided to heat the air after it had been cooled and had passed through the separator, and he hit upon a novel scheme for doing this. His plan was to utilize the heat radiated from the discharge pipe of the ammonia compressor. He took down a section of the 4-in. discharge line and slipped a length of 8-in. pipe over it. The 8-in. jacket pipe was flanged and fitted with a stuffing box at each end.

With this arrangement the engineer can send the air through the annular heating space in the jacket pipe, during cold weather, while in warm weather he can send the air directly to the agitators. A. J. DIXON.

St. Louis, Mo.

[Discussion of Mr. Dixon's plan is invited. It would appear that the engineer cited was not thoroughly familiar with the question of vapor pressure. At any temperature air will carry a certain amount of water vapor. If the air is cooled, the vapor pressure is reduced and some of the water vapor congeals and forms into drops of water. As an example, at a dry-bulb temperature of 80 deg. F. there will be 11 grains of moisture per cubic foot of space. If, now, the temperature be cooled to 40 deg., the air will be able to carry only 3 grains of water vapor per cubic feet. If the air is reheated as outlined by Mr. Dixon to, say, 80 deg., no more moisture will settle out, and if the air be confined in a pipe line no more moisture will be taken up. On passing down the agitator pipe in the ice can, the temperature will drop to, say, 35 deg., at which temperature only 2.4 grains of water can be carried as a vapor. There is 3 grains in the air so the 0.6 grain will settle out and freeze in the vertical pipe.

Suppose the engineer had not heated the air after cooling. The 3 grains of vapor which the air can carry at 40 deg. will be carried over to the ice can and, being

cooled on the way down the agitator pipe, will freeze as before.

If the air velocity is high, it might be possible for the air and moisture to be carried out of the pipe into the can before the freezing point is reached, but this is extremely doubtful.

The engineer would have been on the right track if he had put brine through the double-pipe cooler and reduced the air to practically can temperature. Then all the moisture would have been thrown down and the agitator pipe would not have frozen, since little or no temperature drop would have occurred at the nozzle.—Editor.]

What Is the Matter with These Diesel Diagrams

In Figs. 1 and 2 are two indicator diagrams taken from an 18 x 24 four-stroke-cycle vertical four-cylinder oil engine operating at 200 r.p.m. The spring scale is 200 lb.

I would like to have some *Power* reader explain why, in Fig. 1, the card is so sharp, why the combustion line drops so fast and why the diagram is thick at the lower end.

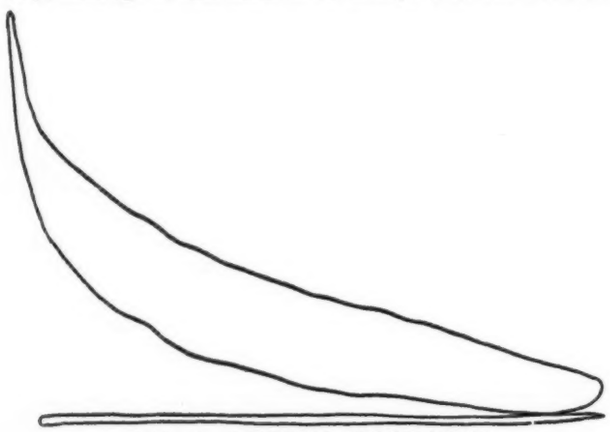


FIG. 1. DIAGRAM SHOWS PEAK

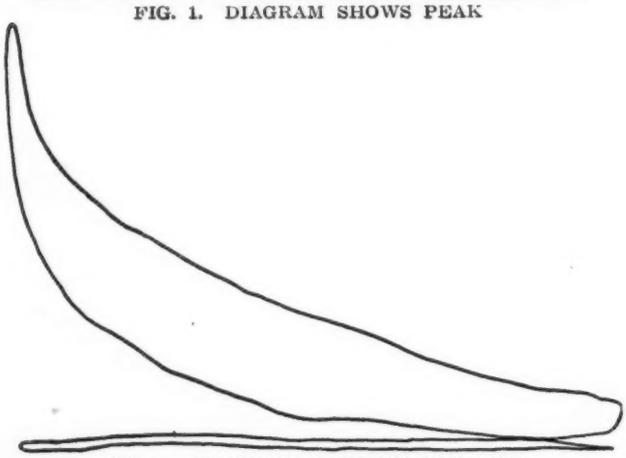


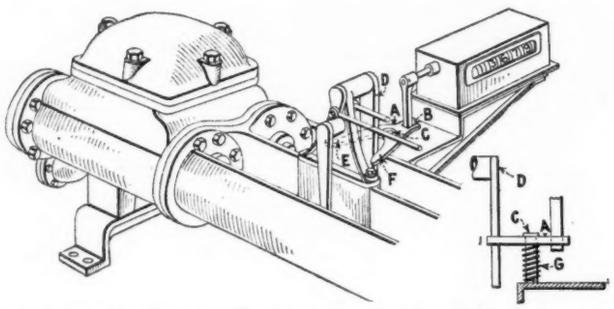
FIG. 2. PECULIAR COMPRESSION LINE

Turning to Fig. 2, which was taken after the fuel-valve opening was set 6 deg. later (making it 1 deg. ahead of dead center), why does the compression line slope to the right at its top? Why are all the lines in both diagrams so wavy? Was this due to cylinder conditions or was something wrong with the indicator?

Kansas City, Mo. L. HAVERY.

Revolution Counter Rig for a Pump

After trying several devices for the correct operation of a revolution counter for a duplex pump or other reciprocating unit where the stroke is not constant, with unsatisfactory results, I devised a motion that gives the correct count, although the piston stroke may be only a short distance past the center either way.



COUNTER ATTACHMENT THAT GAVE CORRECT READING

To do this, put the long arm of the bell crank of the pump in a vertical position. Then, by using the rule of three, calculate and make a rocker, such as shown at *A* in the illustration, where *B* is the counter-arm slot, *C* the fulcrum hole, *D* the pump arm and *E* and *F* forks against which the pump arm pushes. The rocker *A* is placed in a horizontal position so that the pump arm when in a vertical position will be central with regard to the rocker. The counter is then set so that the counter arm engages in the slot *B*.

It is obvious that when the pump arm *D* moves the rocker fork *F* out of its path, the fork *E* is left in the path of the pump arm *D* on its return stroke. The spring *G* is used to keep the rocker tight so that it will not be jarred out of position when the arms are not engaged.

Struthers, Ohio. JOHN V. EMERY.

His First Night Job

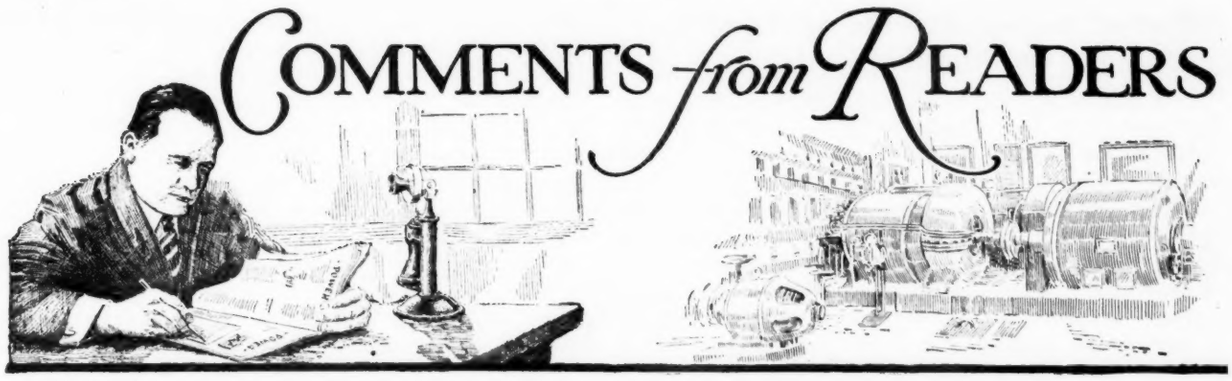
My first job was given me by a friend, when I needed a job mighty bad. I was only seventeen. It was the first night I had ever been up all night, and it was a brand new experience for me.

A steam-driven simplex pump had been installed in a coal mine and was supplied with steam by a small vertical boiler located on the surface, in a ravine through which ran a small stream, which furnished feed water.

My friend was superintendent of construction and had charge of this particular job. One night in June he took me over after everything was working satisfactorily and stayed with me a short time to see that I got a fair start for the night; then he left me, cautioning me to keep the steam pressure up to 100 lb. It was dark and lonesome. I kept my lantern hanging outside and I stayed in the shadows and in this way passed through the night. Oh, how glad I was to see the first sign of the coming day! This brings me to the climax of my first night's work.

I had kept the steam pressure up and everything was going nicely, when I decided to stretch out for a minute's rest. About three hours later I awakened to find my friend standing over me. With a smile he said, "Your steam pressure is a little low." Sure enough, the fire was out and the boiler cold.

Anderson, Ind. FREDERICK L. RAY.



Comments on Breakdown of 30,000-Kilowatt Turbine

In the Feb. 21 issue is a letter from N. W. Akimoff, referring to the accident to the Philadelphia Electric Co.'s turbine.[1] Commenting on the discussion by the writer, in the Dec. 27 issue, he strongly implies that the methods described, or some equivalent, may be found in the works of Föppl, Lord Rayleigh and Love. This statement does not carry conviction, and he should give a specific reference in order to establish his contention. The writer has carefully searched existing literature and has been unable to find a method of determining the natural period of vibration of irregular bodies which can be used for this purpose.

The difficulty in this problem is not a question of mathematics, as he intimates, but the utter impossibility of using the conveniences of mathematical analysis in the problem, since the elastic properties of irregular bodies cannot be reduced to the algebraic expression that is necessary for the reduction of equations to a concrete solution. Mr. Akimoff also states that the solution of this problem offered by the writer is incorrect. This method has been used for many years in the practical design of machinery and has been found uniformly correct in actual experience.

I am afraid that if this problem were begun with a partial differential equation, its solution would remain a partial differential equation. Perhaps, however, Mr. Akimoff can explain the necessity for the partial differential which is not apparent to me.

Is there anything significant in the discovery that one turbine in motion caused the wheels of a standing machine to vibrate, other than to confirm the explanation of Professor Moore, which has been seconded by the writer? This observation simply means that the natural period of the wheel in the standing machine coincided with the speed of rotation of the machine in service and, if applicable to the turbine in question, is exactly the sum total of the explanation we have offered, and with which Mr. Akimoff seems to disagree.

His letter indicates that he probably has in mind that such an accident could have been prevented had the unit in question been in better mechanical balance. While this is certainly a desirable condition for any machine, he does not establish that lack of balance was the cause of the accident. Assuming that he is right in stating that the machine was running rough prior to the accident, it is possible that the roughness noted was not due to lack of balance, which presumably would have been corrected by balance weights, but because of the axial vibration of the wheel which ultimately broke, and if propagated by steam this could not be corrected by balancing.

With reference to the two types of vibration having diameters or circles as the stationary nodes, he is entirely correct, and a full description of these two styles of vibration may be found in some elementary books on physics. What he means by "gyroscopic reactions" in a rotating turbine wheel is entirely beyond the writer's comprehension. The phenomenon usually referred to as "gyroscopic reaction" consists in the inter-relation between rotation of the center line of a revolving mass and the corresponding torque at right angles to the plane of this rotation developed by the revolving mass.

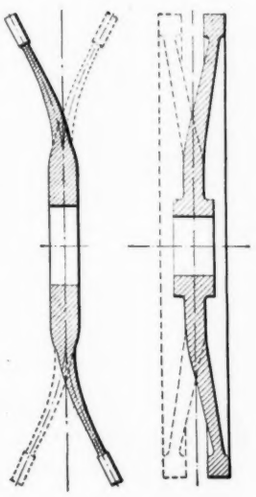


FIG. 1 FIG. 2

Fig. 1 — Vibration at critical speed. Fig. 2— Fluttering of turbine disc.

No turbine mounted in a power plant is arranged for rotation "as if mounted on a" turntable. In consequence the shaft center line cannot be rotated, and therefore no "gyroscopic reaction" can be developed in the wheel. On the other hand, if a gyroscope were mounted on a turntable and rotated, the deflection of the wheel resulting is exactly the reverse of Mr. Akimoff's assertion and is of the type which is shown in Fig. 1.

The vibration of Fig. 2, so strongly advocated by him, however, is difficult to imagine, since it requires axially directed periodic propagating forces for its maintenance, whose origin is obscure, and such vibration would be actively damped in a turbine by the resulting oscillating flux of steam from one side of the wheel to the other. It is immaterial which of these two types of vibration are present, since the natural period or number of vibrations per second which a turbine wheel will have is seldom very different for either type and they are calculated by identical methods and would have similar effect in stressing the disc. Anyone sufficiently interested may readily reproduce all these phenomena by using a small model made from a soft rubber disc and a spindle, which is more readily understood than a partial differential equation.

C. H. SMOOT.

New York City.

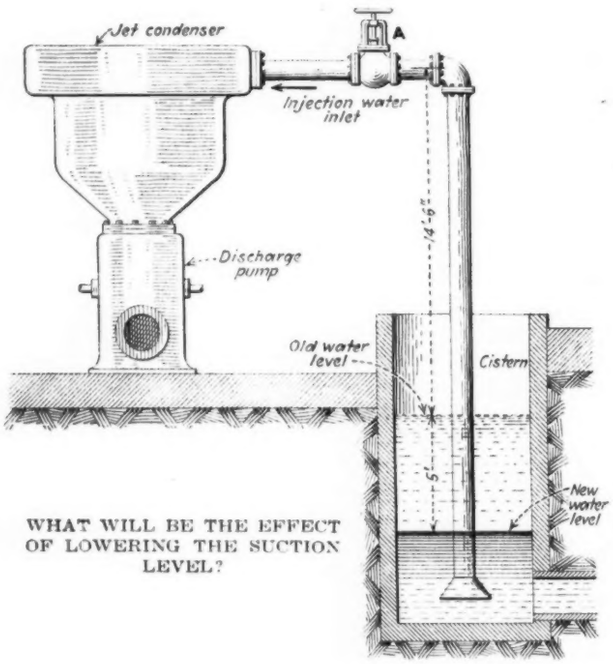
[In the article by N. W. Akimoff in the Feb. 21 issue, the Figs. 2 and 3 were inadvertently transposed.—Editor.]

[1] See *Power*, Nov. 22, 1921.

Effect of Suction Level of Jet Condenser on Pumping Cost

With reference to the illustration and the letter on the effect of suction level of jet condenser on pumping cost in the March 7 issue, Mr. Ray evidently has a definite problem.

Without making any changes in the discharge line from the jet-condenser pump, he is right, because no change in the power consumption will be caused by a change in the injection-water lift. The power consumption in this type of condenser depends on the quantity of water being handled, the efficiency of the pump and the total head against which the pump must operate.



WHAT WILL BE THE EFFECT OF LOWERING THE SUCTION LEVEL?

This total head includes the vacuum on the suction side added to the delivered pressure on the discharge side.

From the illustration it appears that with the change in the water level it might be possible to obtain some benefit from siphon action on the pump discharge. Unfortunately, all the conditions are not shown in the illustration, so that whether or not this is possible or practical cannot be stated without further information.

There is a practical angle to this problem which should not be overlooked. The condenser operation will not be so reliable with the increased suction level on the injection pipe for two reasons.

First, a similar change in vacuum will affect the flow of water, and if the vacuum in the condenser drops below 17 in., due to a slug of air or other causes, water will cease to flow in the condenser.

The total force available for causing the liquid to flow from the cistern into the condenser is equal to the difference between the atmospheric pressure and the absolute pressure existing in the condenser. If the vacuum in the condenser is "high," this difference approaches the total maximum of approximately 14.7 lb. per sq.in. This difference in pressure is all that is available:

a. To give the water the velocity head to enter the pipe.

b. To lift the water through the vertical elevation necessary to raise it from the cistern level to the level of the jet-condenser inlet.

c. To overcome the pipe friction through the straight pipe and the friction of the fittings (the short-radius elbow indicated in the sketch is not particularly good practice on account of the high friction).

d. To overcome the friction through control valve *A* (part of this friction may be necessary in order to limit the quantity of water flowing).

e. To force the water through the spray nozzles in the jet-condenser head.

This introduces the second practical reason which should be kept in mind: The amount of water that can be admitted to the condenser depends to a large extent upon the pressure loss through the spray nozzles. If the difference in head is reduced greatly, the amount of water will be reduced greatly. It is barely possible in the particular case in mind, that valve *A* shown in the illustration might be opened enough to reduce the friction loss an amount equal to the 5-ft. difference in the cistern level.

As a practical problem the decision in this case should also be based upon the character of service that this condenser supplies. If an interruption in the vacuum would cause little annoyance or loss, and the benefits derived from lowering the water level are worth while, the proposed arrangement might be used, but if the condenser is used on central-station service on which many important customers depend, the proposed change would not be recommended. J. C. HOBBS.

Pittsburgh, Pa.

In the March 7 issue, without going over the whole situation again, I believe Mr. Ray to be correct in stating that the lowering of the level of injection water will not increase the cost of pumping. In fact it is possible that just the opposite may result.

For instance, we will assume that under the old conditions the removal pump is discharging the water back to the same level from which the condenser takes it. Assuming the discharge to be sealed, then the total head on the removing pump will be approximately thirty feet, or the equivalent of the vacuum effect on the suction of the pump. The discharge head will be approximately zero. If the water levels of injection and discharge both are then lowered five feet to the new level and the discharge pipe from the pump sealed again, the total head on the pump will be about twenty-five feet, made up as follows:

Head due to vacuum effect	30 ft.
Minus head due to level of discharge tunnel	5 ft.
Total head	25 ft.

As no more work is required to get the water into the condenser, there will be the saving due to the difference in work the removal pump is called upon to do.

New York City. J. J. CONWAY.

If the throttle valve on the water inlet is open at all times and the vacuum is constant, then the flow of water into the condenser will be decreased when the level in cistern is lowered, because the quality of water passed through the condenser nozzles depends on the effective head on these nozzles. This head is varied by the change of vacuum and the change of total suction head, made up of velocity and entry head, friction, static suction head, and siphonic losses.

Increasing the static suction head is a serious thing to do from an operating point of view. First, since the

quantity of water is decreased, the vacuum will fall; secondly, if a serious leakage of air occurs, the vacuum would fall, the water quantity would fall off as the square root of the pressure on the condenser nozzles; thirdly, should some of the nozzles become plugged, the vacuum would be lost altogether; and lastly, the condenser would not be able to operate at full load with warm injection water and not at all under some conditions of hot injection water, as encountered in summer with spray ponds.

JOHN KIRGAN.

Phillipsburg, N. J.

Mr. Ray brings up a question that cannot be disposed of by so simple an analysis as that upon which he bases his conclusion. Whether this conclusion is correct depends upon the following items: The normal vacuum carried; the amount of opening of the valve *A* under best operation; the absolute pressure on each side of the valve *A*; the capacity of the water-injection pipe; the temperature of the cooling water, and the tightness of joints against air leakage.

Neglecting the consideration of air mixed with the exhaust steam, the least cost of pumping, for a given vacuum, will be obtained when the minimum amount of circulating water is used to condense the steam. The temperature of the mixed condensate and cooling water then closely corresponds to the saturation temperature of the exhaust steam.

First, let us suppose, under the old conditions, there is a 24-in. vacuum (3 lb. per sq.in. abs.) in the condenser head, the valve *A* to be wide open and the injection water pipe to be of generous size. The absolute pressure at the elbow (top of vertical pipe) will then be atmospheric pressure minus the pressure of the water column 14 ft. 6 in. high, or 14.7 — 6.3 = 8.4 lb. per sq.in. The water will then shoot into the condenser under a pressure difference of 8.4 minus 3 lb., or 5.4 lb., which large pressure difference may cause much greater water flow than necessary and force the pump to do much more work than is necessary. Under such circumstances, to lower the suction level would, by lessening the absolute pressure at the elbow, diminish the water flow; the pump could be slowed down and the cost of pumping lessened. This is only because the operation was bad in the first place.

On the other hand, assume the same conditions as above except that the valve *A* has been partly closed because the operator recognized the fact that too much water was flowing. This will throttle the water flow and possibly bring it to that value required for the ideal condition mentioned. If, now, the water level in the cistern is lowered, the valve *A* may be opened farther, to allow the same flow as before, and the cost of pumping will be the same.

It may be, however, that in the original installation the injection water pipe was of such small size that the resulting friction and velocity losses reduced the pressure at the elbow to an extent requiring valve *A* to be wide open. It might also be that these conditions allowed the most economical water flow. If the cistern level is lowered, there will be less pressure at the elbow, and water flow will be lessened. This will cause the vacuum to fall, if the pump speed is unchanged. But the pump may be speeded up, in which case a higher vacuum may be obtained in the condenser chamber than before, and then the necessary minimum water flow will be restored.

All of the foregoing conclusions may be modified through consideration of the actual vacuum carried, the effect of increased air infiltration with increased vacuum and the temperature of the cooling water.

Mr. Ray's statement "that the vacuum in the condenser is maintained for a purpose entirely independent of the lifting of the condensing water" is not to the point. He might equally well say that the draft on a boiler furnace is maintained for a purpose entirely independent of the effect of air infiltration through cracks in the boiler setting. And equally well, he might conclude that there is no loss through such infiltration. On the face of it his analysis seems to state that an additional amount of work can be done without extra effort. But no closed heat power system can perform additional work without receiving additional energy. This additional energy may, as discussed, have been waste energy before the change; that is, waste because of throttling water pressure past valve *A*.

A vacuum is *not* "maintained for the purpose of condensing steam," but, instead, the steam is condensed in order to maintain the vacuum.

Baltimore, Md. JULIAN C. SMALLWOOD.

The level of the water in the cold well can be lowered to 19 ft. 6 in. and will not increase the pumping costs of the removal pump, for the reason that the removal pump does not lift the water, the function of the same being to remove injection water and condensate.

The function of the vacuum pump is to remove the air out of the condenser. After that is removed and vacuum maintained, the injection water will flow and can flow into the condenser only after the air has been removed and a vacuum of 20 in. maintained. The water then will be forced into the condenser by atmospheric pressure.

The important point is not the pumping costs, but whether it will be possible to obtain the necessary vacuum equal to the height of the suction lift with a turbine outfit. Obtaining a high vacuum on a standing machine is getting to be a difficult proposition owing to the turbine manufacturers issuing a series of "don't do this and don't do that."

For instance, to obtain a high vacuum on a standing outfit, you would have to seal the turbine glands with steam and this is objectionable on account of distorting the packing, also heating the shaft in the gland. With the water-sealing glands it would be difficult to obtain a 20-in. vacuum until the turbine was up to at least one-half speed, when the impeller would build up sufficient pressure to overcome air leakage.

This important question is not one of pumping costs, but rather, can you obtain a vacuum at all times high enough to cause the water to flow continuously while on load? If not, be prepared to use a forced injection to get your vacuum back if it should fail. In making these comments, I am assuming that this is a turbine outfit with no valve between the turbine and condenser.

Elizabeth, N. J. O. W. FOWLER.

The effect of lowering the suction level will be on the water supply only, and if this is not affected the power for pumping will not be, as the power required depends only on the volume of water and the vacuum in the condenser out of which it must be pumped. If, however, the additional five-foot lift on the suction side is more than the resistance now interposed by

valve A to give the proper regulation of supply and level in the condenser, the vacuum will not lift so much water as formerly against the vertical head and resistance of the piping and condenser nozzles; hence, the volume of water to be pumped and the power to pump it will be decreased.

A simple way to try this is to select a time when the condenser is not in use and produce a vacuum and water circulation; then by throttling the air pump, reduce the vacuum until the valve A is wide open and the normal volume of circulation water is being maintained. If this vacuum is approximately 4.5 in. less than that normally carried the resistance interposed by the valve A is greater than the increase in suction head, and there will be no decrease in the volume of water supplied. If not, the suction pipe or nozzles must be enlarged or the condensing-water supply will be deficient and the vacuum impaired.

We are now operating a condenser of this type and find that with the regulating valve wide open we can maintain full water supply with a 20-in. vacuum in the condenser. As we normally operate at 28 to 28.5 in., this would give us a margin of 3 to 3.5 in. if we made a change similar to Mr. Ray's.

As a general thing condenser manufacturers specify for such equipment that the suction head, including pipe friction, shall not exceed 18 ft., so that the proposed change exceeds this materially, and unless a supply of water under pressure is available it may be difficult in warm weather to start up the condenser.

New Haven, Conn. H. D. FISHER.

[Later solutions have been received from the following: F. A. Buchanan, Gloucester, Mass.; L. P. Allen, Johnston City, N. Y.; E. Meyer, Stamford, Conn.; G. Severance, South Lincoln, Me. These agree substantially with Mr. Ray's theory that no more power would be required.—Editor.]

Heat Transferred in Ammonia Condenser

In the Jan. 24 issue of *Power*, page 156, R. G. C. asks how much heat will be transmitted by a square foot of condenser surface between ammonia vapor at a temperature of 280 deg. and cooling water at a temperature of 60 deg. The answer given took the form of an equation

$$H = K\left(\frac{t_1 - t_2}{2}\right)$$

in which H is the heat transferred per square foot of surface per hour, K is the coefficient of heat transmission of pipe per square foot per degree temperature difference per hour, t_1, the temperature of the ammonia vapor was given as 280 deg. F., and t_2, the temperature of the cooling water intake was 60 deg. F.

Substituting a value of 30 for K, the solution of this equation gives 3,300 B.t.u. per sq.ft. per hour as the value for H.

The fundamental equation for the transfer of heat between the ammonia vapor and the cooling water, through the walls of the pipe is

$$H = K \times T_m \times S_m$$

where H is the total heat in B.t.u. transferred per hour; K is the coefficient of heat transmission from ammonia vapor to cooling water, through the walls of the pipe, expressed in B.t.u. per square foot per degree difference in temperature per hour; T_m is the mean temperature difference between the vapor and cooling water, and S_m is the mean, or effective, surface of the pipe in square feet.

Since the heat transferred per square foot per hour is desired, S_m will become unity, in this case. The equation will then be

$$H = K \times T_m$$

where H is the heat transferred per square foot per hour.

The use of 30 as a value for K would indicate that the thermal conductivity of the pipe metal had been confused with the coefficient of heat transmission from the ammonia, through the walls of the pipe, to the cooling water. The two are considerably different, owing to the presence of film formation on the inside and outside of the pipe. The curve in Fig. 1 shows values of K which should be expected for different velocities of cooling water. These have been verified by actual tests and may be used with confidence.

The velocity of the cooling water may be found by weighing it, finding the rate of discharge in pounds per minute. This may be changed to cubic feet per minute by multiplying by the conversion factor 0.016. The cubic feet per minute discharge divided by the inside area of the pipe in square feet will give the velocity of the water in feet per minute.

In stating the conditions, it has been assumed that the cooling water remains at a constant temperature of 60 deg. Of course this is not strictly true, as the temperature must necessarily rise if heat is abstracted

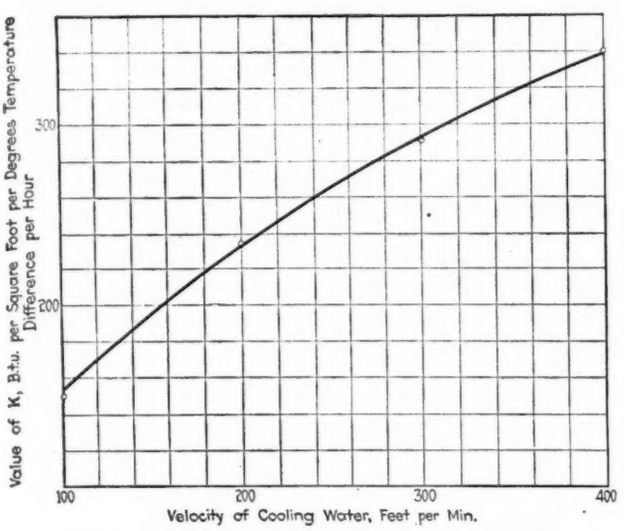


FIG. 1. COEFFICIENT OF HEAT TRANSMISSION

from the ammonia vapor. The temperature of the cooling water should be recorded as it enters the ammonia vapor and again as it leaves it. The mean temperature difference between the ammonia vapor and the cooling water is not the arithmetical mean, but the logarithmic mean, which may be found by the formula,

$$T_m = \frac{A_1 - A_2}{nat.\ log\left(\frac{A_1}{A_2}\right)}$$

where A_1 is the temperature of ammonia vapor minus temperature of inlet water, and A_2 is the temperature of ammonia vapor minus the temperature of outlet water.

Ammonia vapor at 280 deg. indicates considerable superheat. Since no data are furnished of the pressure in the condenser, it is necessary to assume the pressure, which we will say is 185 lb. gage. This saturation temperature is then 96 deg. The diagram, Fig. 2, indicates how the heat is extracted by the cooling water, which first flows through the condensing section and then

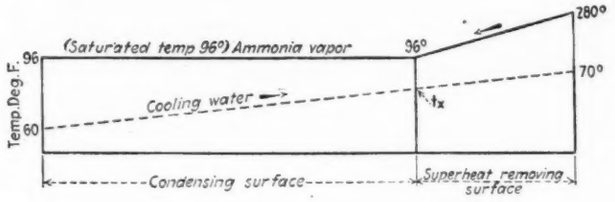


FIG. 2. CONDENSER TEMPERATURES

through the superheat-removing section of the condenser.

It is necessary to find the logarithmic mean temperature difference for both the condensing and superheat-removing sections, because this difference varies so widely in each. In order to do this, it is necessary to know the temperature of the cooling water at the beginning of the superheat-removing section. This is indicated in Fig. 2 as T_x, and may be easily and accurately calculated by noting the per cent of the total heat removed which is taken away by the cooling water in the condensing section. In this case the total heat of the superheated ammonia at 280 deg. is 676 B.t.u. The heat of the liquid at the saturated temperature is 72 B.t.u. Hence the total heat removed is 676 — 72 = 604 B.t.u. Of this, the condensing section removes the latent heat of vaporization, which at 185 lb. gage is 488 B.t.u., or $\frac{488}{604} = 81$ per cent of the total. Hence 81 per cent of the total temperature rise (70 — 60 = 10 deg.) occurs in the condensing section, from which $T_x = 60$ deg. + 8.1 deg. = 68.1 deg.

The mean temperature may now be computed for the two sections. For the condensing section,

$$T_m\ (cond.) = \frac{(96-60)-(96-68)}{\log_e\left(\frac{96-60}{96-68}\right)} = 31.9\ deg.$$

For the superheat-removing section,

$$T_m\ (super.) = \frac{(280-70)-(96-68)}{\log_e\left(\frac{280-70}{96-68}\right)} = 90.5\ \text{deg.}$$

Now, as has been previously noted, 81 per cent of the heat is removed by the condensing section, and accordingly 19 per cent will be carried off by the superheat-removing section. The mean value of heat transfer for the whole condenser, which we have taken as 150, is computed by the formula,

$$K_m = \frac{\frac{H_s}{T_{ms}} + \frac{H_c}{T_{mc}}}{S_m}$$

where H_s and H_c represent the heat carried away by the superheat-removing and condensing surfaces respectively, and T_{ms} and T_{mc} the mean temperature differences respectively. S_m is the total effective surface in both the superheat and saturated sections.

This formula may be expressed for our use as

$$K_m = \frac{\frac{H(total)\,(.19)}{90.5} + \frac{H(total)\,(.81)}{31.9}}{S_m}$$

or

$$K_m = \frac{H(total)\left[\frac{.19}{90.5} + \frac{.81}{31.9}\right]}{S_m}$$

It is seen by inspection of the foregoing formula that T_m for the whole condenser is equal to

$$\frac{1}{\left[\frac{.19}{90.5} + \frac{.81}{31.9}\right]} = 36.4\ deg.$$

The amount of heat transferred per square foot may then be found, since

$$\frac{H}{S_m} = K_m T_m = 150 \times 36.4 = 5420\ B.t.u.\ per\ sq.ft.$$

Cambridge, Mass. ROSCOE H. HYSOM.

Why Do Piston Rings Break?

Referring to O. Hilliary's troubles with breaking packing rings, mentioned on page 351 of the Feb. 28 issue, I would say that water entering the cylinder causes the rings to break. I experienced considerable trouble with an 8 x 48-in. steam drag saw which broke packing rings quite often, because the operators did not drain out the water on starting. The trouble was overcome by making sections of the packing rings and inserting spring-steel bands behind the sections. For the 8-in. diameter cylinder two steel bands of No. 20 B. & S. gage were inserted behind each packing ring.

In spite of the fact that superheated steam is used on this engine, it is suspected that water gets to it from a dead-end steam main that is excessively large for the amount of service required; or water may come over from boilers through superheaters at certain times, caused by priming due to excess concentration.

A daily analysis of a water sample taken from each boiler may reveal a surprise if the practice is not already being followed in the case in question. Excess concentration is likely to run very high if the lime-and-soda process of water softening is used and daily blowing out of boilers is not practiced. E. G. HOPPE.

Cairo, Ill.

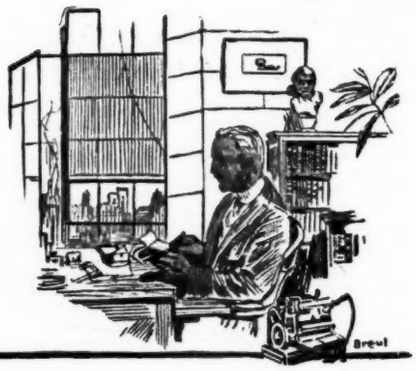
Number 80 Still in M. E. B. A.

In the Jan. 31, 1922, issue of *Power* a statement was made in the report of the annual convention of the Marine Engineers' Beneficial Association, that the charter of Ocean Lodge, No. 80, of New York City, had been revoked. Will you publish the following statement:

A ruling was made by national president Brown, following the last national convention of the M. E. B. A., that the charter of M. E. B. A. No. 80 had been canceled, but inasmuch as the convention had voted not to cancel the charter of M. E. B. A. No. 80, an order was issued by Justice Bijur of the Supreme Court of the State of New York, restraining Mr. Brown and any of the national officials of the M. E. B. A. from interfering with the charter or business of M. E. B. A. No. 80. Therefore, there are two subordinate associations of the M. E. B. A. in the port of New York, No. 33 and the steamship engineers' association, No. 80.

New York City. B. L. TODD,
Manager, M. E. B. No. 80.

QUESTIONS AND ANSWERS



Relative Capacity of Forced- and Induced-Draft Fans

What would be the relative size or capacity of a fan required for mechanical draft or induced draft?

J. N.

The relative capacities required depend on the conditions. For a given boiler the volume of gas passing through an induced-draft fan is much greater than would be handled by a forced-draft fan, depending on the draft pressure, the temperature of the gases, and to some extent on their analyses. Under conditions of average plants the fan capacities required for induced draft are about 50 per cent greater than for forced draft.

Exposure of Girth-Seam Lap to Overheating

Is the lap of a girth seam of a return-tubular boiler more endangered to burning if it comes over the fire or over the bridge wall, or just behind the bridge wall?

W. N. C.

The greatest effect of the fire on the bottom of a return-tubular boiler takes place immediately behind the bridge wall, and when a seam is located at that point there is danger of the lap becoming burned. The action of the heated gases is less severe, and therefore injurious overheating of the lap is less likely, when the joint comes over the bridge wall, and there is still less danger when the joint is over the furnace forward of the bridge wall.

Heat Loss from Moisture in Coal

What effect has moisture in coal on its heat value and how is the loss determined? W. R. N.

Water in coal must be evaporated and superheated and escapes at the flue temperature at atmospheric pressure. Heat is thus rejected to the chimney and therefore any moisture in coal, in addition to being a dead weight, results in waste of heat. The heat can be determined by the formula,

$$\textit{Loss in B.t.u. per pound of dry coal} = M\,[212 - t + 970.4 + 0.47\,(T - 212)]$$

in which

M = Pounds of moisture per lb. of dry coal;
t = Room temperature;
T = Temperature of the flue gases;
970.4 = Latent heat of evaporation f. and a. 212° F.;
0.47 = Mean specific heat of superheated steam.

For example, assume a coal of 13,000 B.t.u. (dry basis) 8 per cent moisture, 60 deg. F. temperature of coal as fired and 550 deg. F. stack temperature. Then by substituting in the formula, the heat loss is found to be $0.08\,[212 - 60 + 970.4 + 0.47\,(550 - 212) =$ 102.5 B.t.u. per lb. of coal, or, $102.5 \times 100 \div 13{,}000$ = about 0.8 of one per cent of the heat in the dry coal, from which it may be noted that with small percentages of moisture the loss amounts to approximately 1 per cent for each 10 per cent of moisture.

Formula for Safe Working Pressure for H. R. T. Boiler

What is the formula for determining the safe working pressure of a return-tubular boiler? W. F. P.

The safe working pressure is the greatest estimated pressure per square inch that can be carried by the boiler without bringing greater stress on any part than would be permissible with application of the assumed factor of safety for the boiler. The safe working pressure therefore depends on the most vulnerable element.

In a return-tubular boiler, with girth seams single-riveted and the heads properly stayed, the safe working pressure usually depends on the longitudinal joint in the shell. The formula for the safe working pressure, as limited by a longitudinal joint of a cylindrical shell or drum, is:

$$P = \frac{TS \times t \times E}{R \times FS},$$

in which

P = Safe working pressure, pounds per square inch;
TS = Ultimate tensile strength of shell plates, pounds per square inch of cross section;
t = Thickness of plate, in weakest course, inches;
E = Efficiency of longitudinal joint;
R = Inside radius of the weakest course of the shell or drum, inches;
FS = Factor of safety, which for new construction usually must be not less than 5.

Preventing Corrosion of Cooling Coils in Transformers

We have been experiencing trouble with the cooling coils in oil-insulated water-cooled transformers corroding to the extent of allowing the cooling water to leak into the transformers. This corrosion takes place only where the brass tubes leave the inside of the cast-iron cover. The insulation on the tubes at the point where the corrosion occurred was found to be in bad condition. Is this trouble one that is common, and what can be suggested as the cause and remedy of the difficulty?

B. A. B.

The cause of the trouble is one that is not prevalent, although a few cases of this kind have occurred. Two sources can be looked to for the cause of the trouble, although the corrosion was due to small traces of acid

in the transformer. One of the sources of acid may be due to traces of copper sulphate being present in the tubes because of imperfections in the manufacturing process. This sulphate, combining with the water vapor that forms on the cold end of the tubes, produces sulphuric acid, which would attack not only the tube, but also the insulation on the tube, especially if this material was a cotton fabric or contained cotton. This acid could also set up an electrolytic action between the brass tube and the iron cover, much the same as electrolytic action is set up around where an iron fence post is held in concrete or stone with lead. In this case it is common to see iron posts eaten away entirely at the base owing to the galvanic action between the iron and lead due to the salts in the water from the sidewalks. Another source of acid is in shellac, in which small amounts of acetic acid are present. This acid is very volatile and could easily condense on the cold tubes.

Probably the best remedy for this trouble is to lag the tubes from below the oil level right up to where they go through the cover. This lagging should be a good heat insulator so as to prevent condensation on the tubes, which is the real cause of the trouble. This lagging may consist of several layers of asbestos cord and over this three or four layers of tape. If the tube were covered with a good moisture- and acid-resisting insulating compound, such as Bakelite, it might be helpful.

Interpretation of Safety-Valve Formula

Where the formula for the size of safety valve for a boiler is $A = W \times 70 \times 11 \div P$, in which A equals the area of direct spring-loaded safety valve in square inches per square foot of grate surface, W equals the weight of water in pounds evaporated per square foot of grate surface per second, and P equals the pressure (absolute) at which the safety valve is set to blow, how would the formula be adapted to the use of oil as fuel?

W. E. T.

As in the original formula A represents the area of safety valve per square foot of grate surface, then

$$\textit{Total area of safety valve} = \textit{number of sq.ft. of grate} \times W \times 70 \times 11 \div P$$

But as square feet of grate $\times W =$ total weight of water evaporated per second, and as evaporation per pound of oil $\times$ number of pounds of oil burned per second would amount to the same thing, then for oil burning, the formula for area of safety valve would become,

$$\textit{Area of spring-loaded safety valve in square inches} = \textit{evaporation per pound of oil} \times \textit{number of pounds of oil burned per second} \times 70 \times 11 \div P.$$

Foaming and Priming From Soda Ash

Does soda ash in a boiler-feed water cause foaming and priming?

W. C.

Soda ash of itself will not cause the trouble. When foaming occurs with the use of soda ash, it usually will be found that proper provision is not made for removal of the precipitates formed by its reaction with the scale forming impurities in the water, and priming and foaming result from the water becoming befouled with impurities. Foaming may also occur from introduction of soda in an open heater that receives animal oil with the exhaust. The mixture of this oil with water containing soda ash results in a soapy emulsion that is favorable to foaming which is made worse by the presence of silt or precipitation of lime and magnesia compounds.

Disconnecting Keyed Piston Rod from Crosshead

How can a piston rod be loosened from a crosshead when the connection is made by driving a tapered key?

W. W.

When the piston rod makes a taper fit in the crosshead and the parts are brought together by driving a tapered key in the direction indicated by the arrow, Fig. 1, the rod is drawn into the crosshead by contact

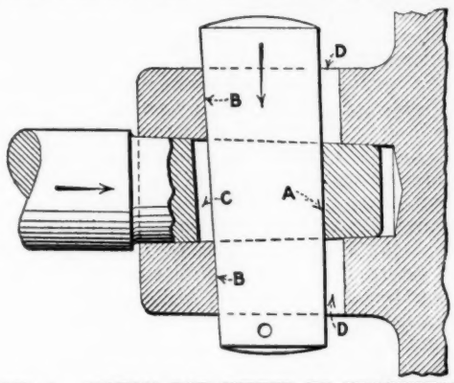


FIG. 1. PISTON ROD KEYED TO CROSSHEAD

of the edge A of the key with the front end of a slot near the end of the rod, while the opposite edge of the key, BB, presses against the back end of the slots in the hub of the crosshead. It is to be noted that, to permit of draft from the wedging action of the key, there must be a clearance space C in the back end of the slot in the rod and similar clearances D and D between the key and the front end of the slots in the crosshead hub.

To drive the rod out of the crosshead, the bearings and clearances must be reversed. Remove the regular

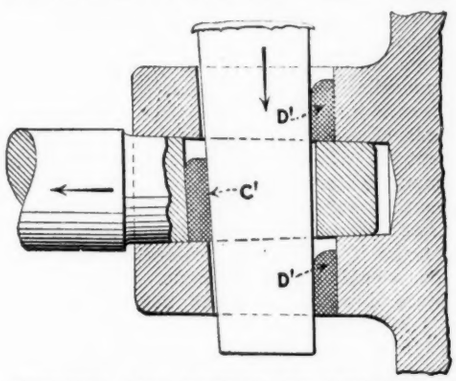


FIG. 2. FILLERS AND TEMPORARY KEY FOR DRIVING PISTON ROD OUT OF CROSSHEAD

key, and as shown in Fig. 2, place fillers C' and D' in the clearance spaces C and D shown in Fig. 1, having these fillers a little wider than the clearances obtained when the regular key is in place. Then, by driving a narrower key that bears on the fillers, the rod will be driven out of its taper fit in the crosshead.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Hydro-Electric Conference Held in Philadelphia

Turbines of 55,000 Hp. and a Hydro-Electric Plant of 600,000 Hp. Capacity—A New Method of Measuring the Flow of Water in Pipe Lines—Radical Changes in Turbine Design—Other Features in Water-Power Development

MANY features of water-power development were discussed at the two sessions of the Hydro-Electric Conference held at the Manufacturers Club, Philadelphia, March 21. This conference was held under the joint auspices of the Engineers Club of Philadelphia and Philadelphia sections of the American Institute of Electrical Engineers, American Society of Civil Engineers and American Society of Mechanical Engineers in co-operation with the Power Division of the American Society of Mechanical Engineers.

The conference, the second to be held in Philadelphia, was opened with a short address by William F. James, president of the Engineers Club of Philadelphia, who expressed a desire to see the practice of holding a hydro-electric conference each year in Philadelphia continued, to discuss the many features involved in water-power development. Mr. James turned the meeting over to Dr. Arthur M. Green, Jr., who presided at both the afternoon and evening sessions.

Utilization of Water Resources

A paper, "The Relation of Water Resources to the Country's Development," was presented by Nathan Clifford Grover, dealing with the many uses of water. The speaker pointed out that the development of water power requires a consideration of water for uses other than for power purposes. He expressed the opinion that probably in many cases the use of water for power purposes is one of the least important, since not infrequently the use of water for irrigation, transportation and municipal uses ranks first. The importance of each of these uses will depend to a large extent upon the locality; where in an arid district irrigation might have first place, around large cities the use of water for municipal purposes would rank in first importance. In the coal-mine region where large power plants are built, one of the important uses of water is for condensing purposes.

The speaker emphasized the important part that the development of electrical transmission has played in the development of water power. It is now economical to develop many water-power resources that it would have been impossible to utilize without long-distance electrical transmission. A transmission line built near a small water site may make it economical to develop this power where otherwise it would not be advisable. Such systems as the St. Lawrence development could not be considered if it were not for long-distance electrical transmission and interconnection of large systems. In the Appalachian coal region there are few sites where steam plants of 100,000 kw. can be built because of limited condensing water.

Largest Hydraulic Turbine in Operation

In his paper, "The 55,000-Hp. Turbines for the Queenston-Chippewa Development," Frank H. Rogers described one of the turbines in the Queenston power house of the Hydro-electric Power Commission of Ontario. These units are the highest-powered hydraulic turbines ever built. In the first installation there will be five machines and each unit is designed to develop 55,000 hp. under a head of 305 ft. at a speed of 187½ r.p.m. Owing to the great size of the turbines and the high head under which they operate, a new design of casing was adopted to prevent excessive stresses and eliminate leakage. To develop and maintain high efficiency special designs of draft tubes, guide vanes and runners are used. An improved type of operating gear is incorporated to obtain the best possible speed regulation and to protect the guide vanes from damage due to foreign obstruction.

The great amount of power developed by a single unit makes the question of positive control at all times and continuous operation of prime consideration. For these purposes a special control stand is provided for each unit, containing the necessary indicators and control valves so that a single operator can from one position not only observe the performance of the turbine and all its facilities, but can also start up, adjust the load and shut down the turbine in the course of regular operation, or shut the turbine gate or the main penstock valve if required in an emergency. The total weight of these units is about two million pounds and the thrust bearing is designed to carry a load of one million pounds. The first unit was put in operation Dec. 28, and a second unit has recently gone into service. When the plant is completed, it will have a total capacity of about 600,000 kw.

Measuring Water in Pipe Lines

"Salt Velocity Method of Measuring Water in Pipe Lines," presented by Charles M. Allen, describes a new method of measuring the flow of water in pipe lines. Salt is introduced into the water at the upstream end of the pipe, and its passage by one or two stations down stream is detected by the use of electrodes in the water and an ammeter connected to an electric circuit. Stop-watches or chronographs are used to determine the time of first appearance of salt, of maximum density or conductivity and the final appearance. From experiments thus far conducted, the time of maximum density appears to give the true time of mean velocity when compared to quantities measured by a weir and venturi meter. The time of maximum density also checks with the color-velocity method. This method has the advantage over the salt-solution method in that it only requires small quantities of salt, where the salt-solution method requires large quantities. Tests made so far have been confined to a 400-ft. steel-pipe line 40 in. diameter, and a 1,400 ft. wood-stave conduit 13 ft. diameter. Although the development work in this method has not been carried to a point where definite conclusions can be drawn, series of tests have shown it to be accurate within less than 1 per cent.

In a paper, "The Correlation of Momentum and Energy Changes in Steady Flow with Varying Velocity and the Application of the Former to Problems of Unsteady Flow of Surges, in Open Channels," R. D. Johnson shows that nature makes eddies in flowing water, according to a definite law, in order that the energy losses thus produced will have a magnitude exactly sufficient to permit of the simultaneous operation of two known laws—conservation of momentum and conservation of energy. The magnitude of surges caused in open canals by varying demands of water at the lower end, as in the case of a water-power development with a turbine drawing water from the canal, is treated of. Although the paper is a mathematical treatment of the subject, the author in his conclusions gives a very interesting detailed study of surges in a hypothetical canal for loads thrown off, in which, by the use of a series of pictures, the various currents and waves in the canal are shown, which takes place until the water reaches a normal condition.

The last paper of the afternoon session, "The Hydraulic Turbine in Evolution," by H. B. Taylor and L. F. Moody, was presented by Mr. Moody, who called attention to a number of radical changes in turbine design at various periods in its development. One of these was when the first turbines were built for the Niagara Falls plant, where large single-runner units were designed to be connected directly to the generator shaft. After this came a period of development when multiple runners were used directly connected to electric generators. This practice was later discontinued and a return made to the single-runner type. At the present time the turbine is in another transition; namely, from the slow-speed to the high-speed runner. These new designs make it possible to operate turbine runners at practically double the speed formerly used for a given head.

Electric-Power Industry in Healthy Condition

At the informal dinner held in the ballroom of the Bellevue-Stratford Hotel, Samuel Insull addressed the conference on the "Public Utility Industries in the United States." He emphasized the growth and the importance of the public utilities—that is, electric light and power companies, telephone companies and electric traction companies—by saying that they represented an investment of twelve and a half billions of dollars and had a yearly income of about two and a half billion dollars. These figures show how conservatively these utilities are capitalized.

The speaker said that experience during the war showed that there is no necessity of having such large reserve capacity in power stations as formerly thought necessary. Reducing the reserve capacity reduces the cost of money per kilowatt of maximum demand. The electric utilities are now reaching the saturation of the dollars invested as never before.

In power produced by central stations, 62 per cent is steam and 38 per cent water. In the speaker's opinion it is very doubtful if this ratio will change very materially, even if a rapid development in water power is being made. This is due to the fact that where the large industrial centers are, the water power to be developed is small, consequently rapid development in the steam plants will be required to meet the demand. The only way that a material change could be made in this ratio is by the shifting of the industrial centers to where the large water-power developments are. Interconnections of electrical systems possess, in the speaker's opinion, great possibilities in the way of increasing the load factor. In the large cities in the East with their dense populations, the load factor for 6 systems averages 43 per cent, where on the Pacific Coast, with systems interconnected the load factor is 57 per cent, which gives a difference in the use of the generation and transmission equipment of over 30 per cent. This means that capital is 30 per cent cheaper than in the average individual large system.

In closing, the speaker said that he considered the electric-power business to be in a marvelous condition considering what it has gone through for the last six years, and he hoped that the methods that have proved so sound in the past will continue to exist; namely, privately owned and privately operated utilities regulated by government authority.

The first paper presented at the evening session was "Distribution of Hydro-electric Power in Ontario from the Standpoint of the Municipal Customer," by P. B. Yates. The speaker pointed out that the Hydro-electric Power Commission of Ontario was formed to supply the needs in the industrial districts of Ontario for the advantages of cheap electrical power. The development of this desire through legislation designed to meet the difficulties encountered has resulted in the present system fed from Niagara Falls and several other systems supplied from other sources throughout the province, making the Commission's power systems the largest in the world. Mr. Yates went into various details of methods of determining the cost of power supplied to each municipality, the operation of the municipal systems by local commissions, and the standard systems of accounting, rate making and operation. It was shown that the power bill for lighting the average home was about $1.64 per month, less 10 per cent for prompt payment. A 750-watt water heater could be supplied with power at $21 per year.

Three papers were presented on the Queenston-Chippewa power development by H. G. Acres, chief hydraulic engineer, E. T. K. Brandon, electrical engineer, and T. H. Hogg, assistant hydro-electric engineer of the Hydro-Electric Power Commission of Ontario. These papers describe the various engineering features of the Chippewa-Queenston development. Articles describing this installation will be published in early issues of *Power*.

Portable Conveyor Handles Coal and Ash at Low Cost

By A. MacLean*

Five years' operation without a cent for repair parts is the record of our 18-in. by 36-ft. portable conveyor. It averages only three hours' operation so that the cost for power is around 10 cents a day. The conveyor is used in the coal-storage yard for handling coal and ashes and has proved a big labor saver. Using three shovelers, the average cost per day to operate is $6.06, or 13.16c. per ton of coal or ashes handled.

The coal comes in on a siding, is dumped through the hopper bottoms of the cars and loaded by men onto the conveyor, which either raises the coal to small dump cars feeding the hoppers in the boiler room or piles it for storage in the yard. In the latter case, when the coal is to be reclaimed, the conveyor again handles it to the dump cars, so that some of the coal is handled twice. The plant is using approximately 900 tons of coal a month in the summer, and 1,200 tons a month in the winter; In addition, the conveyor handles the ashes, amounting to about 2,000 tons a year. These are hauled from the boiler room on the small dump cars to the storage pile, from which the conveyor loads them into trucks or freight cars.

OPERATING COST OF 18-IN. PORTABLE CONVEYOR

Plant	Mechanical Rubber Co.
Make of Conveyor	Barber-Greene
Excluding Labor	
Depreciation, $900 ÷ 10 yr	$90.00
*Average interest at 6 per cent $= \frac{11}{10} \times \frac{\$900 \times 0.06}{2}$	29.70
Maintenance and repair, estimated	50.00
Power—9 kw.-hr. per day @ $0.01 per kw.-hr. × 300 days	27.00
Total annual operating cost (excluding labor)	$196.70
Cost per day = $196.70 ÷ 300 days	$0.6556
Cost per ton = $0.6556 ÷ 46 tons	$0.0142
Including Labor	
Cost of conveyor per day	$0.6556
Cost of labor per day, 3 men 3 hours per day at $0.60 per hour	5.40
Total daily operating cost (including labor)	$6.0556
Cost per ton = $6.0556 ÷ 46	$0.1316

*Allowing for interest credited on depreciation fund.

On the average the conveyor can do all of this work in three hours' operation per day. Usually, it requires three men to keep it loaded to capacity. As these men have other work to do about the plant, they are charged against the conveyor only when they are actually working with it. With three men loading 46 tons of coal or ashes are easily handled in three hours.

The conveyor is driven by a 3-hp. motor, and during operation consumes about 3 kw. per hour. There have been no repairs in five years, except a few adjustments in the motor. The original belt is still in use, although soon it must be replaced. The actual operating cost of the conveyor, without labor, is $196.70 per year, or $0.6556 per day operated. Adding the labor of the three men usually employed with the conveyor, the cost is $6.0556 per day. The handling cost per ton is 1.42c. without labor, and 13.16c. with labor.

*Engineer, Mechanical Rubber Co., Chicago.

Test on Still Steam-Oil Engine

A test recently made on a Still steam-oil engine, conducted by H. R. Sankey as reported in *The Engineer*, is reproduced here. The Still engine used in the test is a one-cylinder unit. The upper end of the cylinder is the oil-burning portion, while the lower end under the piston is the steam cylinder. In the commercial units it is the intention to employ three cylinders arranged with one cylinder carrying high-pressure steam which exhaus' into the other two cylinders. This gives three single-acting oil cylinders, one high-pressure and two low-pressure steam cylinders. In the experimental engine a separate steam engine was connected to the crankshaft to act as the high-pressure steam portion. This engine was jacketed by the exhaust gases from the oil cylinder, but not sufficiently to make it thermally as efficient as is to be expected in the commercial engine. The steam supply is secured from the oil-engine jacket and exhaust supplemented by a separate boiler.

TABLE I.

	Over-load	Full Load	Three-Quarter Load	Half Load
Dimensions of engine—				
Main cylinder:				
Stroke	36 in.			
Bore	22 in. diam.			
Piston rod	6¼ in. diam.			
Auxiliary high-pressure cylinder:				
Stroke	22 in.			
Bore	14 in. diam.			
Maximum pressures in combustion cylinder, lb. sq.in.	615	595	600	585
Boiler pressures, lb. abs.	114	112	111	111
Exhaust of engine (inches mercury)	26.5	26.9	26.8	25.9
Temperatures (deg. F.) Exhaust gases, boiler inlet	756	678	579	494
Feed water	78	76	79	78
Exhaust gases to atmosphere	156	164	181	181
Steam in boiler	347	346	345	345
Weight of steam passing through engine (pounds per hour)	1,025	892	583	355
M.e.p. referred to combustion cylinder—				
Combustion	89.53	81.37	71.27	56.63
Steam, high-pressure	5.10	3.86	2.07	1.12
Steam, low-pressure	6.46	6.14	4.75	2.72
Total	101.1	91.4	78.1	60.5
Revolutions per minute	129.3	125.0	111.3	97.4
I.h.p. combustion cylinder	400.5	351.5	274.1	190.6
I.h.p., total	451.7	394.7	300.3	203.5
Brake load (pounds)	1,500	1,380	1,125	850
Hp. absorbed by brake	387.9	345.0	250.4	165.6
Brake efficiency, per cent	85.9	87.4	83.4	81.4
Hp. used in scavenging	20.7	17.7	13.5	6.5
Effective horsepower	367.2	327.3	236.9	159.1
Ratio, effective hp. to total i. hp., per cent	81.3	82.9	78.9	78.2
Total oil consumption per hour (pounds)	147.2	126.7	90.9	61.0
Oil consumption per combustion i. hp. per hour (pounds)	0.367	0.360	0.332	0.320
Oil consumption per effective hp. per hour (pounds)	0.401	0.387	0.384	0.383
Thermal efficiency I. hp. per cent	38.4	39.2	42.5	44.1
Thermal efficiency effective hp., per cent	35.2	36.4	36.7	36.8
B.t.u. per minute per I. hp.	110.4	108.2	99.8	96.2
B.t.u. per effective b. hp.	120.6	116.4	115.5	115.2
Lubricating oil per hour, lb.	0.6	0.6	0.6	0.6
Smoke	Slight haze	None	None	None

The essential results of the test are given in Tables I and II, while Table III shows the results to be expected from a three-cylinder engine.

Attention is called to the following facts brought out by the tests:

a. The remarkable fuel economy and the corresponding high thermal efficiency.

b. The small change in this economy from half to full load.

c. The capability of considerable overload at a comparatively small increase in fuel consumption per effective horsepower.

d. The high brake efficiency for a high-compression oil engine.

e. The low consumption of lubricating oil.

f. The low temperature of the exhaust gases on final discharge.

g. The entire absence of smoke, except a slight haze at overload.

At full load the indicated horsepower developed in the combustion cylinder was 351.5, the oil consumption was 126.7 lb. per hour, and since the lower calorific value of the oil was 18,053 b.t.u. per pound, the thermal efficiency was 39.2 per cent. The thermal efficiencies for the other loads are given in Table II.

These thermal efficiencies have been compared with those of an ideal oil engine working between the same temperatur limits as the actual engine at the various loads. The temperature limits have been obtained by the use of a temperature-entropy chart, and the thermal efficiencies calculated, taking account of the variation of specific heat.

TABLE II

	Over-load	Full Load	Three-Quarter Load	Half Load
Thermal efficiency of ideal engine	50.0	50.8	55.0	51.6
Thermal efficiency of actual engine	38.4	39.2	42.5	44.1
Efficiency ratio	76.8	77.2	83.3	85.4

At overload, 1,025 lb. per hour of steam (saturated) passed through the high-pressure and low-pressure cylinders. In the case of the high-pressure cylinder the indicated horsepower developed was 22.8, corresponding to a feed of 45 lb. per indicated horsepower hour. The admission pressure was 111 lb. abs. per in. and the exhaust 22 lb. absolute, and for these steam conditions the feed of the Rankine engine is 21.5 lb. per indicated horsepower-hour. Hence the efficiency ratio is 47.8 per cent, which is distinctly low for a high-pressure cylinder.

For the low-pressure cylinder the following are the corresponding figures: I.hp. developed, 28.9; feed, 35.5 per i.hp.-hour; Rankine feed, 17.1 lb. per i.hp.-hour; efficiency ratio, 48 per cent.

Usually, the efficiency ratio of the low-pressure cylinder of a condensing engine does not exceed 40 to 45 per cent, owing mainly to the loss caused by "cutting off the toe of the diagram." The higher value of 48 per cent in this case is due to the heat transmitted from the combustion cylinder.

It cannot be doubted that in the actual marine engine of the type which will be built for commercial use, where the high-pressure cylinder is at the bottom of a combustion cylinder, so that the steam receives heat during admission and expansion, the efficiency ratio of this high-pressure cylinder would be at least 90 per cent. As to the low-pressure cylinder, an efficiency ratio of 90 per cent can be expected if the back pressure be increased to the release pressure by passing its exhaust into the low-pressure portion of a mixed-pressure turbine provided for driving the scavenge air blower. This turbine should be supplied with a small amount of steam direct from the boiler to make up the power required by the blower and for regulating purposes.

In the experimental engine tested, a certain amount of the exhaust gases was used for jacketing the high-pressure cylinder (auxiliary engine). In the three-cylinder unit this waste of exhaust gases would not occur and a corresponding additional weight of steam would be produced in the boiler.

The mechanical losses of the auxiliary steam engine were included when determining the horsepower absorbed by the brake, but in a three-cylinder unit they would not occur.

TABLE III—ECONOMICAL RESULTS THAT MAY BE EXPECTED IN A THREE-CYLINDER UNIT WHEN THE STEAM IS EXHAUSTED INTO A LOW-PRESSURE TURBINE DRIVING THE AIR SCAVENGE BLOWER

	Over-load	Full Load	Three-Quarter Load	Half Load
Total i.hp.	463	404	302	206
Brake hp.	404	359	256	172
Brake efficiency, per cent	87.2	88.8	84.6	83.2
Oil per brake hp. (pounds per hour)	0.364	0.353	0.355	0.355
Thermal efficiency per brake hp., per cent	38.7	39.9	39.7	39.7
B.t.u. per minute per brake hp.	109.6	106.3	106.8	106.8

The auxiliary engine is single-acting, and judging from general experience, its merchanical efficiency at 40 lb. m.e.p. would be about 90 per cent, which works out to a loss of horsepower equivalent to 0.0342 × revolutions per minute. Being a single-acting engine, the loss is practically constant at all loads, and this horsepower can therefore be added to the effective horsepower.

The calculations to ascertain what improvement in economy may be expected, based on the foregoing, are given in Table III.

News in the Field of Power

Coal Operators Challenged To Show Actual Costs

On the eve of the coal strike, the United Mine Workers of America have issued an interesting statement that challenges the operators to make public their actual cost of producing coal and the profits that they have made. The challenge is in answer to a recent statement of George H. Cushing, managing director of the National Wholesale Coal Association, in which he said:

"On last January 12, the average spot price of coal, mine run basis, f.o.b. mines was $2.13 per ton. Of the $2.13 received by the operator, 67.8 per cent went to labor."

The statement of the United Mine Workers says: "If Mr. McAuliffe's figures were correct, which we do not admit, they would prove that the total cost of producing a ton of coal was only $1.44. In other words, the mine workers received only $1.44 for producing a ton of coal that sold at retail at $8 to $10. Surely, the public will not say the miner is the one that is gouging the coal consumer. . . . These operators say the only way to reduce the price of coal is to cut the miners' wages. The miners challenge them to make public, without camouflage, the actual figures on their cost of production and the actual figures on the profits which they are making out of their business. These figures, honestly stated, would show that someone else, and not the miners, are to blame for the high prices of coal to the consumer."

St. Louis To Have Six-Million-Dollar Plant

A steam-power plant of an initial capacity of 60,000 kw. and probable ultimate capacity of 240,000 kw. is to be put up on the Illinois side of the Mississippi River, about four miles from the electrical center of St. Louis and just south of the city limits. Title to about fifty acres for a site for the plant has been taken by Louis H. Egan, president of the Union Electric Light and Power Co., of St. Louis.

The name of the new plant will be the Cahokia Steam Power Plant. The first two units will be of 30,000 kw. capacity each, operating at 60 cycles, 13,200 volts. The steam pressure will be 300 lb. at the throttle. The estimated cost of the plant is put at over $6,000,000.

The demand for electric power in St. Louis and vicinity is reported to be nearing the point where it will be greater than the available capacity in the large hydro-electric plants of the Mississippi River Power Co. at Keokuk, Iowa, and in the Union Electric Light and Power Co.'s Ashley St. steam plant.

The engineers and constructors in charge of the project are McClellan & Junkersfeld, Inc., 45 William St., New York City. Mr. McClellan is the president of the American Institute of Electrical Engineers, and he and Mr. Junkersfeld have only recently incorporated.

Muscle Shoals Hearings Over, Kahn Opposed to Ford

Hearings before the Military Affairs Committee of the House of Representatives on the disposal of the Government project at Muscle Shoals have been concluded. The committee does not expect to begin the consideration of the matter in executive session until after its return from Muscle Shoals. All but two of the members of the Military Affairs Committee expected to be able to accompany the party of Senators that was to have left Washington March 25 on a special train to visit Muscle Shoals.

It is understood that Representative Kahn, the chairman of the Military Affairs Committee, and whose influence with the committee is a dominating one, is opposed to the acceptance of the Ford offer. He is understood to have reached this conclusion after having heard all the evidence laid before his committee. He entered the hearings, however, rather favorably disposed toward the acceptance of Mr. Ford's offer.

The Republican steering committee of the House is also understood to be opposed to the Ford offer, but a new element has been injected into the situation by the attitude of the Republican Congressional committee. This committee is said to be convinced that failure to accept the Ford offer will bring serious political consequences at the congressional elections this fall. As a result, it is understood that an effort will be made to influence the House to accept the Ford offer, particularly if it can be amended so as to come under the Federal Water Power Act, and provided a certain portion of the power will be made available for public-utility purposes.

Brazilian Centennial Exposition Is Coming

In addition to the international expositions mentioned in a recent editorial in *Power*, there will be held in Rio de Janeiro, from Sept. 7 of this year to March 31 of 1923, an exposition to celebrate the one hundredth anniversary of the Independence of Brazil. The sum of $1,000,000 has been set aside by Congress for purposes of American participation, and will be used for the construction of an official exposition building, in which only exhibits prepared by the various departments of the United States Government will be located. The building will be substantial and permanent in its character, and will be used after the exposition as a home for the American Embassy.

In connection with the exposition there will be held an International Congress of Engineers, probably in September, the proposed subjects for discussion being The Utilization of Fuel Resources, The Best Utilization of Water Power, Recent Advances in Irrigation Methods, The Elimination of Waste through the Standardization of Supplies for Agricultural and Industrial Purposes, Coal as a Factor in Industrial Development, Essentials of a National Railroad Policy, Inter-continental Engineering Co-operation, Port Development, Terminal Facilities, The Iron and Steel Industry.

The participation of American manufacturers as exhibitors is invited, and information as to details may be obtained through the Bureau of Foreign and Domestic Commerce at Washington.

Power Legislation Passed by New York State Legislature

Three important water-power development measures were passed by the New York State Legislature before it adjourned on March 17.

One was the Ferris amendment to the constitution, permitting the legislature to provide for use of not exceeding three per cent of forest preserve lands for development of waterpower and rights of way for electric transmission lines, all of which are declared to be public uses, and permitting lease of water power for terms not exceeding fifty years. It will be necessary to repass this amendment within the next two years before it can go to the people for ratification. It differs from a similar amendment passed by the legislature of 1920 in that the 1920 amendment limited the term of lease to ten years.

The other two measures passed by the legislature are the Robinson water power bills, both of which are in the hands of the Governor. Both measures were the subject of considerable debate in the closing hours of the legislature. The million-dollar appropriation made by one of the Robinson bills (No. 213) for developing water power at the Crescent and Vischers Ferry dams was particularly criticized as being only one-sixth of the ultimate cost.

The other Robinson bill (No. 212), relating to water-power development regulations, was amended so much in the closing hours that it increased from 15 to 29 pages.

Subdivision 2 of section 616 of this measure was amended in relation to

payment of annual charge by licensee by adding "in other cases, except where the state has no proprietary interest, an equitable annual charge may be made, in determining which the commission shall give consideration to the cost of producing power by others in competition with the licensee."

Section 617 was amended by limiting to one the number of preliminary permits that may be issued and outstanding for the same project.

Section 618 was so amended as to allow the power commission to disregard some provisions of the law in the case of a project of less than 100-hp. or minor parts of a larger project.

By the passage of the Ferris amendment to the constitution and the Robinson amendments to the conservation law in relation to the licensing of water power, the state of New York has adopted a definite policy of water power development, through private capital and long term leases; through its passage of the Robinson so-called "hydro-electric" measure, it has started an experiment that in the end may run into many millions of dollars.

Recent Water Power Applications

The application of the Louisville Gas and Electric Company, covering the proposed development of power at Dam No. 41, on the Ohio River, has been received by the Federal Power Commission. The project is expected to develop 25,000 hp., and it is the plan of the gas and electric company to install equipment capable of generating 75,000 hp. The application is made on the probability that the Government will increase the height of the dam by six feet. This matter is being studied by the Corps of Engineers, but it has not been determined finally.

The St. Cloud Public Service Company, of St. Cloud, Minnesota, has applied for a preliminary permit covering the proposed development of 10,000 hp. at Rigby, two miles above Clearwater. It is planned to install 15,000 hp. for public utility purposes.

Stevenson To Help Government in Standardization Work

The American Engineering Standards Committee has designated A. A. Stevenson, its retiring chairman, as a special representative to work with the Department of Commerce in the activities of its Division of Simplified Practice. The division was organized late in 1921, and is now actively co-operating with a number of industries in an effort to cut down the excessive variety of products that has been found to be the cause of so great an amount of waste.

Mr. Stevenson is a past-president of the American Society for Testing Materials, is widely known for his work in behalf of standardization and is vice-president in charge of manufacture of the Standard Steel Works Co., a subsidiary of the Baldwin Locomotive Works. His designation as the representative of the committee was made upon request of Mr. Hoover himself.

The American Engineering Standards Committee was organized a little more than two years ago, and the present results of its activities are that more than a hundred national organizations are co-operating in the movement. Eighteen standards have been formally approved and nationally recognized, and more than fifty others are being prepared. The committee is maintained jointly by five departments of the Federal Government, nine national engineering societies and fifteen national industrial associations.

The committee reports that there are at this time similar standardizing bodies in thirteen foreign countries, those in England and Germany being the most active.

A. S. M. E. Completes Code for Miniature Boilers

The A.S.M.E. subcommittee appointed to formulate a code for miniature boilers has completed its work, and the Council has ordered the Code printed for public inspection. It is expected that a few proof copies will soon be available for the use of those who are directly concerned with the subject. Miniature boilers, as defined in this code, embrace fired pressure vessels that do not exceed the following limits: 16 in. inside diameter of shell, 42 in. inside length of shell, 20 sq.ft. total heating surface, 100 lb. per sq.in. working pressure. Further information may be secured from C. W. Obert, secretary to the A.S.M.E. Boiler Code Committee, 29 West 39th St., New York City.

New Publications

Thermal Expansion of Nickel, Monel Metal, Stellite, Stainless Steel and Aluminum. Scientific Paper No. 426 of the Bureau of Standards. Single copies free from the Bureau, additional copies at 10c. from the Superintendent of Documents, Government Printing Office, Washington, D. C.

Tables and curves are given showing the thermal expansion of 29 samples from room temperature to 600 deg. C., except for stainless steel, which was heated to 900 deg. C. The increasing use of these metals makes the results obtained by this investigation of especial value.

Economics of Electrical Distribution. By P. O. Reyneau and H. P. Seelye. Published by the McGraw-Hill Book Co., Inc., New York, 1922. Cloth; 6 x 9 in.; 209 pages; 59 illustrations; 20 tables. Price, $2.50.

The purpose of this book is to present the need for the application of economic principles to the design of distribution systems, to explain the fundamental principles involved, to indicate the types of problems most often encountered and to offer methods of studying such problems and reaching their solution. The book is made up of two parts. The first part (Chapters II to VII) is intended to give working methods for solving electrical-distribution problems economically. The second part is devoted to presenting the application of the methods given in the first part. For convenience a division of the subject has been made into transmission lines, power circuits, lighting circuits, secondaries and underground lines. Under each heading some of the general problems encountered are solved in detail. Special or unusual problems have not been considered, the work for the most part holding to every-day questions met with in practice. One chapter is devoted to general problems which apply to the system as a whole, such as the location of generating stations and substations. Another chapter takes up briefly a few of the problems pertaining to power distribution in industrial plants. In the limited space devoted to the treatment of the subjects it obviously is impossible to cover the whole field of economics as applied to the central-station systems. However, the book contains much that is of interest to those who have to do with electrical power-distribution problems.

Personals

Peter G. Schilling, formerly with H. S. Johannsen, New York City, is now chief engineer for the American Krupp System-Diesel Engine Co., New York.

Robert H. McLain, formerly sales engineer for the power and mining department of the General Electric Co., at Schenectady, N. Y., is now in a similar position with the Maine Electric Co., Portland, Me.

Frank L. Fairbanks, chief engineer of the Quincy Market Cold Storage and Warehouse Co., of Boston, is going down to Georgia to look over some properties for Stone & Webster.

Robert M. Gates has come back into the power-plant field by joining the Superheater Co., 17 East 42nd St., New York City. Before his association with the Lakewood Engineering Co., he was in the power-plant-equipment business as a partner of the Herron Gates Co., Cleveland, Ohio.

Obituary

Dr. Charles W. Waidner, chief physicist of the Bureau of Standards, died at his home in Washington, D. C. on March 10, after a long illness. He was born in Baltimore March 6, 1873, and received his early education in the public and private schools there, after which he was graduated from Johns Hopkins University, later receiving the degree of Ph. D. from that institution. He taught for several years, after which he entered the Bureau of Standards in 1901, where he remained until his death.

His work has been mainly in the fields of temperature and heat measurement and apparatus for the study of thermal phenomena. In recent years his particular problem has been direction of the work on the determination of physical constants used in engineering, especially fireproofing of structural materials and refrigerating engineering.

Society Affairs

Coming Conventions

American Institute of Electrical Engineers, 29 West 39th St., New York City. Spring convention at Chicago, April 19-21.

American Society of Mechanical Engineers, 29 West 39th St., New York City. Spring meeting at Atlanta, Ga., May 8-11.

American Water Works Association, 153 West 71st St., New York City. Annual convention at Philadelphia, May 15-19.

National Electric Light Association, 29 West 39th St., New York City. Annual convention at Atlantic City, N. J., May 15-20.

Metropolitan Section, A. S. M. E., is arranging for a meeting in New York City, April 4, on "Muscle Shoals."

The Cleveland Engineering Society will meet April 4 at the Hotel Winton to hear a talk on "Making Our Writing Effective," by Karl O. Thompson, of the Case School of Applied Science.

Buffalo Section, A. S. M. E., will hold a joint meeting Apr. 5 with the A. I. E. E. H. M. Cushing, of the Buffalo General Electric Co., will give an illustrated lecture on the company's steam station on the River Road, and motion pictures of stoker operation will be shown.

Business Items

The C. W. Hunt Co., Inc., West New Brighton, S. I., N. Y., maker of material handling machinery, has taken over all the sales and engineering work formerly carried on by its subsidiary company, the C. W. Hunt Engineering Corporation, with which it has now consolidated, and has re-established its New York City office at 143 Liberty Street.

The United Machine and Mfg. Co., of Canton, Ohio, has taken over the Harrington Stoker from the James A. Brady Foundry Co., of Chicago, which will now devote its entire attention to its foundry and ash conveyor business.

Joseph S. Webster has organized a sales force to be known as Joseph S. Webster & Co., which will represent the Kingsford Foundry and Machine Works, Oswego, N. Y., in the New England states for the sale of Kingsford-Webster water-tube boilers. Offices will be established in New Haven, Providence and Boston.

Trade Catalogs

Generating Sets—Engberg's Electric and Mechanical Works, St. Joseph, Mich. A 32-page catalog on the Engberg direct-connected steam-engine generator sets, which are made for either 110- or 220-volt service, with capacities of from 1 to 50 kw.

Refillable Fuses—Cote Bros. Manufacturing Corporation, First National Bank Bldg., Chicago, Ill. A short folder on the new "Simplicity Fuse," a plug-type fuse that may be easily and quickly renewed and that has other convenient features.

Refrigerating Machinery — Norwalk Iron Works Co., South Norwalk, Conn. Bulletin No. 7, 15 pp., interesting chiefly because of its description of two new products—a fully assembled refrigerating unit, of either one-half or one-ton capacity, and a new type of simplified condenser. The company makes also a fully assembled quarter-ton refrigerating outfit and ammonia compressors in sizes from one to ten tons.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run bases, f.o.b. mines):

Coal	Market Quoting	March 13, 1922	March 20, 1922
Pool 1,	New York	$2.95	$2.75@3.00
Pocahontas,	Columbus	1.85	1.75@2.00
Clearfield,	Boston	1.95	1.65@2.25
Somerset,	Boston	1.90	1.75@2.00
Pittsburgh,	Pittsburgh	2.15	1.90@2.10
Kanawha,	Columbus	1.60	1.40@1.65
Hocking,	Columbus	1.90	1.65@1.90
Pittsburgh No. 8	Cleveland	1.90	1.85@1.90
Franklin, Ill.,	Chicago	2.50	2.25@2.75
Central, Ill.,	Chicago	2.35	2.25@2.50
Ind. 4th Vein,	Chicago	2.40	2.40@2.50
Standard,	St. Louis	1.85	1.75@1.85
West Ky.,	Louisville	1.85	1.65@1.90
Big Seam,	Birmingham	1.85	1.70@2.00
S. E. Ky.,	Louisville	1.50	1.50@1.65

New York—On Mar. 23, Port Arthur light oil 22@25 deg. Baumé 3⅞c. per gal. 30@35 deg., 4¾c. per gal., f.o.b. Bayonne, N. J.

Chicago—Mar. 8, for 24@28 deg. Baumé, 85@90c. per bbl.; 21@36 deg., 2⅜@2½c. per gal. in tank cars f.o.b. Oklahoma refinery, or freight adjusted.

Pittsburgh—On Mar. 14, f.o.b. refinery, Pennsylvania, 38@40 deg. 4¼c. Kentucky fuel oil, 26@30 deg., 3c. per gal. Western, 24@30 deg., 80c. per bbl.; 32@34 deg., 2⅜c.; 36@38 deg., 3c.; 38@40 deg., 3¼c. per gal.

St. Louis—Mar. 18, prices f.o.b. cars, tank lots; 24@26 deg. Baumé, 90c. per bbl.; 26@28 deg., 95c.; 28@30 deg., $1.10; 32@34 deg., 2⅜c. per gal.

Philadelphia—On Mar. 20, 26@28 deg. Baumé, Oklahoma, 85@90c. per bbl.; 30@34 deg., Oklahoma (group 3) 2⅜@3c. per gal.; 16@20 deg., Seaboard, $1@$1.10 per bbl.

Cincinnati—Mar. 15, for 26@30 deg. Baumé, 4¼c.; Diesel 32@34 deg., 4½c. per gal.; distillate 38@46 deg., 5¼c. per gal.

New Plant Construction

PROPOSED WORK

Ala., Birmingham—The Bd. Educ. will receive bids about April 20 for a 3 story school (2nd unit) on 8th Ave., N. About $500,000. D. O. Whilden, Archt.

Cal., Los Angeles—The Central Inv. Co., c/o Schultz & Weaver, Archts., 633 Pacific Mutual Bldg., received bids for a 14 story, 198 x 320 ft. hotel including a steam heating system on 5th and Olive Sts. from Lang & Bergstrom, Washington Bldg., $4,991,773; Scofield Engrs. & Constr. Co., 728 Pacific Finance Bldg., $5,086,000; and J. Stewart & Co., $5,210,000.

Cal., San Francisco—Capo di Monte, c/o T. P. Ross, Archt., 310 California St., has had plans prepared for a 6 story apartment on Filbert and Hyde Sts. About $400,000.

Conn., Hartford—C. J. Malmfeldt, Archt., 206 Trumbull St., will receive bids until April 1 for a 7 story, 50 x 160 ft. hospital including a steam heating system on Hudson St. for the Hartford Hospital. About $500,000. Noted Feb. 21.

Conn., New Haven—R. W. Foote, Archt., 185 Church St., will receive bids until March 31 for a 12 story, 64 x 90 ft. office building including a steam heating system, 3 boilers, on Church and Court Sts. for A. W. Powell, 177 Church St. About $1,000,000.

Col., Colorado Springs—The Methodist Episcopal Church, E. Hislop, First Methodist Church, Boulder and Nevada Aves., Chn. of committee, has purchased a site on East Boulder, Main, East Willamette and Foote Sts. and plans to build a hospital. First unit to cost, $250,000. Total cost about $1,000,000.

Conn., Hartford—The High School Plant & Bldg. Comn., W. H. Scoville, Chn., will receive bids until April 11 for a 3 story high school including a steam heating and ventilating system on Greenfield and Ridgefield Sts. About $1,500,000. F. I. Cooper Corp., 33 Cornhill, Boston, Mass., Archts.

Ill., Chicago—The Chicago, Burlington & Quincy R.R., 547 West Jackson Blvd., is receiving bids for a 6 story, 300 x 700 ft. freight house including a steam heating system on Canal, Harrison and Polk Sts. About $3,000,000. W. T. Krausch, care of owner, archt.

Ill., East St. Louis—The Union Electric Light and Power Co., 12th and Locust Sts., St. Louis, Mo., L. H. Egan, Pres., has secured a site on the Mississippi River near here for a steam power plant to have an initial capacity of 60,000 kw., ultimate capacity about 240,000 kw. The first 2 units will be 30,000 kw. each, supplied with steam at 300 lbs. pressure at the throttle and will deliver electrical energy at 60 cycles, 13,200 volts. Cost about $6,000,000. McClellan & Junkersfeld, 45 William St., New York City, Engrs.

Ill., Hinsdale—The Village, G. M. Williams, Clk., plans to build a 1 story ice plant. About $25,000. Private plans.

Ind., Bloomington—The Indiana University will receive bids until April 6 for a 3 story, 75 x 185 ft. commerce and finance building. About $350,000. R. F. Daggett, 959 Limcke Annex, Indianapolis, Archt.

Ind., Indianapolis—The Bd. Sanitary Comrs., City Hall, will receive bids until March 30 for furnishing power plant equipment and appurtenances for the Sewage Disposal Plant, including three 400 hp. water tube boilers; three chain grate stokers for boilers; three steam turbine blowers; two steam turbine electric generators, 500 kw. each; five surface condensers; one electric switch board; mechanical equipment; mechanically operated weir gates, etc.

Ind., Indianapolis—The Natl. Military Home, P. G. Froemming, Treas., is receiving bids for a feed water heater and purifier and an engine and generator.

Kan., Holton—The city is having plans prepared for improvements to power plant including the installation of 100 kw. steam uniflow generating unit. About $10,000.

Black & Veatch, Municipal Bldg., Kansas City, Mo., Engrs.

Minn., Duluth—The St. Louis County Bd., Court House, will receive bids until April 14 for a 4 story, 115 x 168 ft. city jail including a steam heating system on West 2nd St. About $392,000. Former bids rejected.

Mo., Joplin—The Miners Ice & Fuel Co., 318 Main St., J. E. Campbell, Mgr., is in the market for a steam driven compressor, about 450 to 500 cu.ft. free air, 110 lb. working air pressure.

N. J., Frenchtown—The council is having plans prepared for extensions to its municipal light and power plant. Causbey Engr. Co., 1011 Chestnut St., Phila., Pa., Engrs.

N. J., Montclair—The Montclair Rink & Ice Co., c/o G. Backhoff, Archt., 9 Clinton St., Newark, is having sketches made for a 1 story, 125 x 250 ft. ice plant and skating rink. About $200,000.

N. Y., LeRoy—The LeRoy Lime & Crushed Stone Co., operating large quarries and a crusher near here, plans to install electric power for operating of the crushing machinery. The LeRoy Hydraulic Electric Co. will run a high power line to the quarry, furnishing 250 hp. The steam plant will be retained by the company for emergency.

N. Y., Mount Morris—The Mount Morris Canning Co., Inc., is in the market for a 60 hp. portable or locomotive type steam boiler.

N. Y., New York—The Bd. of Health, 505 Pearl St., will receive bids until April 3 for furnishing labor and material for erection of a heating plant, boiler, etc., on the grounds of the Willard Parker Hospital, Foot of East 16th St.

N. Y., New York—The Bronx Boosters Inc., J. M. Haffen, Pres., c/o Maynicke & Fanke, Archts., 25 East 26th St., is having preliminary plans prepared for a 10 story, 122 x 236 ft. hotel including a steam heating system on the Grand Concourse between 161st and 162nd Sts. About $750,000.

N. Y., New York—The Comr. Water Supply, Municipal Bldg., will receive bids until March 31 for furnishing installing and connecting a boiler at the 98th St. pumping station. About $14,000. Former bids rejected. Noted March 7.

N. Y., Saranac Lake—The Village Water Bd. and the Paul Smiths Electric Light & Power Co. plans to build a dam at Saranac Lake here. Address C. L. Dickert, Saranac Lake. Engineer not selected.

N. Y., Syracuse—W. M. Wischeroth, 100 Otisco St., is in the market for a boiler to develop about 40 hp. or more, also two or three vats, 4 x 4 x 12 by ¼ in. thick.

N. C., Lexington—The Bd. of Town Comrs., Town Hall, J. A. Leonard, Mayor, will receive bids until April 6 for improvements to waterworks including pumping station, filter house, etc.

Pa., Phila.—R. Beard, Archt., 2120 North Lambert St., is receiving bids for a 1 story, 227 x 288 ft. theatre on 5th and Poplar Sts. for the Maxwell Amusement Co., care of the archt. About $350,000.

Pa., Phila.—The Bd. Educ., 19th and Chestnut Sts. received bids for a 3 story school on 24th and Jackson Sts. from M. Nelody & Sons, 1322 Race St., $566,870; P. H. Kelly Co., 1713 Sansom St., $575,000; heating system from C. E. Monday & Co., 1318 Olive St., $81,762.

Pa., Pittsburgh—R. M. McAnulty, 421 Park Bldg., is in the market for an air compressor, jack hammer and a 15 hp. a.c. motor.

Va., Norfolk—The city, W. H. Taylor, Dir. Pub. Wks., will receive bids until March 31 for switchboard equipment at Lake Prince Dam pumping station. D. A. Decker, Engr.

W. Va., St. Marys—R. W. Russell is in the market for two 125 to 150 hp. vertical gas engines, Bruce-Macbeth type preferred.

W. Va., Wheeling—The Wheeling Machine Products Co., 1920 Main St., E. W. Krause, Treas. and Genl. Mgr., is in the market for one 250 to 300 hp. direct connected steam generator unit, including boiler, Ball, Allis-Chalmers or Ides preferred.

Wis., Madison—The city, H. C. Buser, Clk., is having plans prepared for the installation of a gasoline engine, generator, etc., and equipment for South Side Disposal plant. E. E. Parker, Engr.

Wis., Milwaukee—Cahill & Douglas, Engrs., 217 West Water St., is receiving bids for a 200 hp. motor generator set for Geuder, Paeschke & Frey, 15th and St. Paul Ave., manufacturers of enamel ware.

Wis., Milwaukee—M. A. Weinreis, 241 Lake St., is in the market for 2 air compressors, one 10 x 10 x 10 and one 8 x 8 x 8, also one skeleton hoist and two hoists with boilers.

Wis., Spring Green—The city, A. Schlosser, Clk., will receive bids until April 14 for furnishing and laying water mains here, also 14 valve boxes, 50,000-gal. steel tank on 125-ft. tower, one 300-g.p.m. motor and engine driven pump, etc. About $25,000. W. G. Kirchoffer, Madison, Engr.

Ont., Dunnville—The Pub. Utilities Comn., W. Marshall, will receive bids until March 31 for 1,200,000 gal. waterworks filtration plant with filters, controls and all equipment. About $100,000. E. H. Darling, Home Bank Bldg., Hamilton, Engr. Noted April 5.

Ont., Guelph—The Bd. Educ. will soon receive bids for a 3 story collegiate including a vacuum steam with mechanical ventilation system. About $320,000. G. Hutton, Hamilton Bank Bldg., Hamilton, Archt.

Ont., Saulte Ste. Marie—The city is having plans prepared for the installation of two gasoline engines for direct connection to present pumps in the pumping station. About $20,000. W. L. McFaul, City Hall, Engr.

Ont., Seaforth—The council, J. A. Wilson, Clk., will receive bids until April 8 for a reservoir, pumphouse and elevated tank. About $71,000. James, Proctor & Redfern, 36 Toronto St., Toronto, Engrs.

Que., Montreal—The Canadian Wirebound Boxes, Ltd., 17 Mill St., is in the market for a 40 hp. electric motor.

Que., Montreal—The St. Lawrence Baling Co., 23 St. Francois Xavier St., is in the market for three electric motors, one 7½ hp., one 3½ hp. and one 2 hp.

Que., St. George's—The Provincial Government plans construction of a dam on the Chaudiere River here to develop 2,000 hp. About $800,000.

Samoa, Tutuila—The Bureau of Yards and Docks, Navy Dept., Wash., D. C., will receive bids until April 19 for two 300 ft. radio towers. Spec. 4600. Noted March 14.

CONTRACTS AWARDED

Cal., Fresno—The California Hotel Co., S. N. Griffith, Promotor, has awarded the contract for a 20 story hotel on Mariposa and M Sts. to the Lindgren Co., Monadnock Bldg., San Francisco. About $2,000,000.

Cal., Grossmont—S. L. Studios, San Diego, A. H. Sawyer, Pres., has awarded the contract for a 90 x 200 ft. motion picture producing plant (1st unit), to Miller & Fifield, San Diego. About $40,000. Total cost, $500,000.

Cal., Los Angeles—The Globe A No. 1 Ice Cream Co., c/o W. A. Hertman Constr. Co., 900 Merritt Bldg., archt., has awarded the contract for a 2 story, 148 x 276 ft. ice cream factory on Hill and Jefferson Sts. Contract was awarded to the Architect. Conveyors, refrigeration system and ice making machinery will be installed.

Cal., Los Angeles—The Westinghouse Electric & Mfg. Co., Van Nuys Bldg., has awarded the contract for a 6 story, 170 x 239 ft. warehouse including a steam heating system on 5th and San Pedro Sts. to E. B. Newcomb, Pantages Bldg. Cost plus percentage basis. Noted Jan. 24.

Cal., San Diego—The Bureau of Yards and Docks, Navy Dept., Wash., D. C., has awarded the contract for steam supply system at the Naval Base Training Station to G. E. Engr. Co., 449 West 42nd St., New York City. $168,300. Spec. 4586.

Conn., New Britain—The Amer. Hosiery Co., Park St., will build a 1 story, 40 x 60 ft. boiler house at plant. About $25,000. Work will be done by day labor under supervision of engineers. Noted March 21.

Ill., Chicago—C. R. Gleason, 111 West Washington St., has awarded the contract for a 3 story apartment on Winthrop and Glenlake Aves. to A. E. Johnson, 5356 Belleplain Ave. About $500,000. A steam heating system will be installed.

Ill., Chicago—W. L. Wallen & Sons, 6713 North Clark St., have awarded the contract for a 3 story, 173 x 580 ft. store and apartment bldg. on Estes and Glenwood Ave. to C. B. Johnson & Son, 111 West Washington St. About $2,000,000. A steam heating system will be installed.

Ind., Indianapolis—The Libby Realty Co., c/o Vonneaut, Bohn & Mueller, Archts., 610 Indiana Trust Bldg., has awarded the contract for a 12 story, 77 x 90 ft. office building on Illinois and Washington Sts. to the Bedford Stone & Constr. Co., Fletcher Trust Bldg. About $500,000.

Ind., Whiting—The Bd. Educ. has awarded the contract for a 3 story, 90 x 140 ft. high school and a 1 story, 70 x 90 ft. grade school to English Bros., 419 Lincoln Bldg., Champaign, Ill. About $500,000. A steam heating system will be installed. Noted Feb. 7.

Kan., Salina—The Kansas Wesleyan University, L. K. Bowers, Pres., has awarded the contract for a 4 story, 150 x 202 ft. administration building to Eberhardt Constr. Co., Salina. About $350,000.

Kan., Salina—The Natl. Bank of America, F. Hageman, Pres., has awarded the contract for a 4 story, 50 x 120 ft. bank and office building to the Eberhardt Constr. Co., Salina. About $300,000. Noted Feb. 21.

Mont., Missoula—The State University of Montana Bd. of Examiners has awarded the contract for two 1 story, 45 x 74 ft. heating plants including equipment, one at State University, here, and one at College of Agriculture and Mechanical Arts, Bozeman, to the Northern Plumbing & Heating Co., Billings, $331,627.

N. Y., Brooklyn—The Bd. Educ. has awarded the contract for the installation of a heating and ventilating system in P. S. No. 187 on 12th Ave. to Raesler Htg. Co., 129 Amsterdam Ave., New York City, $78,197, also a heating and ventilating system in P. S. 115 on East 97th St., to T. F. Hall Inc., 405 Lexington Ave., New York City, $44,956.

N. Y., Brooklyn—G. & H. Bldg. Corp., c/o Shampan & Shampan, Archts., 50 Court St., will build a 4 story, 100 x 151 ft. apartment on 15th Ave. and 53rd St. About $250,000. Owner will build by day labor. A steam heating system will be installed.

N. Y., New York—The New York State Realty & Terminal Co., c/o Cross & Cross, Archts., 681 5th Ave., has awarded the contract for a 12 story, 100 x 125 ft. office building at 395 Madison Ave. to Todd, Robertson & Todd, 101 Park Ave. About $1,000,000.

N. Y., New York—Terry & Tench Co., Inc., Grand Central Terminal, has awarded the contract for furnishing and installing gasoline driven generator, power motors, controller, electric lighting, etc., at Troy Cohoes Bridge over the Hudson River, to John R. Proctor, Inc., 120-22 Liberty St.

N. Y., Niagara Falls—The city has awarded the contract for a school to Wright & Kremers, Niagara Falls, $463,200. Heating to the Power Efficiency Corp., White Bldg., Buffalo, $107,900.

N. C., Raleigh—The State Dept. of Agriculture has awarded the contract for a 5 story agriculture building on Edenton and Halifax Sts. to J. E. Beaman, Conmercial Bank Bldg., $312,000. Noted April 5-21.

Pa., Phila.—D. Crawford, Jr., 701 North 63rd St., will build a 10 story, 175 x 220 ft. apartment on 39th and Chestnut Sts. About $3,000,000. Owner will build by day labor. A steam heating system will be installed. Noted Jan. 10.

Pa., Phila.—The Curtis Publishing Co., Independence Sq., has awarded the contract for a 3 story, 145 x 215 ft. printing plant (first unit) on 7th and Sansom Sts. to Doyle & Co., 1519 Sansom St.

Pa., Tarentum—The borough has awarded the contract for removing part of present structure, building 29 x 43 ft. concrete and brick pump house together with foundations for same, also concrete foundation for 2,000,000 gal. steam pump, wood construction roof for 35 x 44 ft. concrete tank; foundation and superstructure to Mars Constr. Co., Mars, pump to Worthington Pump Co., Oliver Bldg., Pittsburgh.

Va., Roanoke—The Viscose Corp. has awarded the contract for the installation of generating unit to the Westinghouse Electric & Mfg. Co., Union Bldg., Pittsburgh, Pa. About $1,000,000.

Wis., Richland Center—The city has awarded the contract for equipment for power plant including engine generator unit, exciter, switchboard and centrifugal pumps to Allis-Chalmers Mfg. Co., West Allis, $17,970.

Wis., Sheboygan—The Bd. Educ., J. Walboord, Supt., 1809 West 5th Ave., has awarded the contract for a 2 story, 278 x 368 ft. high school on 9th and Jefferson Sts. to J. M. Hansen, 113 West Walnut St., Green Bay. About $1,000,000. A steam heating system will be installed.